TRANSACTIONS

OF THE

AMERICAN SOCIETY

OF

MECHANICAL ENGINEERS.

VOL. XXV.

XLVIIIth Meeting, New York, N. Y., 1903.
XLIXth Meeting, Chicago, Ill., 1904.

NEW YORK CITY:
PUBLISHED BY THE SOCIETY,
From the Library Building,
No. 12 West 31st Street.
1904.

Press of J. J. Little & Co.
Astor Place, New York.

East Eng. Lib.,
Mr. W. E. Muggett
9t
6-27-1924

OFFICERS

OF THE

AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

1903-1904,

FORMING THE STATUTORY COUNCIL.

PRESIDENT.

AMBROSE SWASEY .... Cleveland, O.

VICE-PRESIDENTS.

F. H. DANIELS .... Worcester, Mass.
JAMES CHRISTIE .... Philadelphia, Pa.
JOHN R. FREEMAN .... Providence, R. I.

Terms expire at Annual Meeting of 1904.

D. S. JACOBUS .... Hoboken, N. J.
M. L. HOLMAN .... St. Louis, Mo.
WILLIAM J. KEEP .... Detroit, Mich

Terms expire at Annual Meeting of 1905.

MANAGERS.

R. S. MOORE .... San Francisco, Cal.
H. A. GILLIS .... Richmond, Va.
CHAS. H. CORBETT .... Brooklyn, N. Y.

Terms expire at Annual Meeting of 1904.

R. C. McKINNEY .... New York, N. Y.
S. S. WEBBER .... Trenton, N. J.
NEWELL SANDERS .... Chattanooga, Tenn.

Terms expire at Annual Meeting of 1905.

GEORGE I. ROCKWOOD .... Worcester, Mass.
JOHN W. LIEB, JR .... New York, N. Y.
ASA M. MATTICE .... Pittsburg, Pa.

Terms expire at Annual Meeting of 1906.

TREASURER.

WM. H. WILEY .... Nos. 43-45 East 19th St., New York, N. Y.

SECRETARY.

PROF. F. R. HUTTON .... No. 12 West 31st St., New York, N. Y.

HONORARY COUNCILLORS.

PAST PRESIDENTS OF THE SOCIETY.

THURSTON, R. H.	1880—1882	Died Oct. 25, 1903
LEAVITT, E. D.	1882—1883	Cambridge, Mass.
SWEET, JOHN E.	1883—1884	Syracuse, N. Y.
HOLLOWAY, J. F.	1884—1885	Died Sept. 1, 1896
SELLERS, COLEMAN	1885—1886	Philadelphia, Pa.
BABCOCK, GEORGE H.	1886—1887	Died Dec. 16, 1893
SEE, HORACE	1887—1888	
TOWNE, HENRY R.	1888—1889	Stamford, Conn.
SMITH, OBERLIN	1889—1890	Bridgeton, N. J.
HUNT, ROBERT W.	1890—1891	Chicago, Ill.
LORING, CHARLES H.	1891—1892	Brooklyn, N. Y.
COXE, ECKLEY B.	1892—1894	Died May 13, 1895
DAVIS, E. F. C.	1894	Died Aug. 6, 1895
BILLINGS, CHARLES E.*	1895	Hartford, Conn.
FRITZ, JOHN	1895—1896	Bethlehem, Pa.
WARNER, WORCESTER R.	1896—1897	Cleveland, O.
HUNT, CHARLES WALLACE	1897—1898	New York, N. Y.
MELVILLE, GEORGE W.	1898—1899	Philadelphia, Pa.
MORGAN, CHARLES H.	1899—1900	Worcester, Mass.
WELLMAN, S. T.	1900—1901	Cleveland, O.
REYNOLDS, EDWIN	1901—1902	Milwaukee, Wis.
DODGE, JAMES M.	1902—1903	Philadelphia, Pa.

[Note.—According to the Constitution, Article C 27, the five surviving Past Presidents who last held the office shall be members of the Council, with all the rights, privileges and duties of the other members of the council.]

* Unexpired term of E. F. C. Davis.

NOTE.

The considerable bulk of the volume of Transactions has induced the Publication Committee to direct the insertion of a summary of the Society membership in place of the complete list of members which was published in the earlier volumes. The summary attaching to this issue is that which appears in the catalogue of the Society issued with corrections to July 1st, 1904. Reference for the complete list should be made to the "Geographical List" for July, 1904.

FOREIGN COUNTRIES.

	Membership.		Membership.
Africa	20	Holland	1
Australia	4	India	2
Belgium	3	Jamaica, W. I.	1
Canada	31	Japan	4
Central America	1	Mexico	6
China	3	Norway	1
Cuba	2	Russia	5
France	9	South America	7
Germany	7	Sweden	4
Great Britain (England)	44	Switzerland	1
Great Britain (Scotland)	4	Trinidad, B. W. I.	1

Total foreign membership 161

UNITED STATES.

	Membership.		Membership.
Alabama	5	Montana	9
Alaska	1	Nebraska	3
Arizona	1	New Hampshire	15
Arkansas	2	New Jersey	130
California	30	New Mexico	1
Colorado	23	New York	802
Connecticut	111	North Carolina	4
Delaware	14	North Dakota	1
District of Columbia	28	Ohio	188
Georgia	13	Oklahoma	1
Hawaiian Islands	1	Oregon	4
Illinois	164	Pennsylvania	377
Indiana	33	Porto Rico	2
Iowa	3	Rhode Island	48
Kansas	3	South Carolina	3
Kentucky	4	South Dakota	1
Louisiana	11	Tennessee	4
Maine	13	Texas	5
Maryland	35	Utah	2
Massachusetts	249	Vermont	11
Michigan	67	Virginia	25
Minnesota	13	Washington	5
Mississippi	1	West Virginia	8
Missouri	41	Wisconsin	62

Total membership in the United States........2,577

GEOGRAPHICAL SUMMARY.

Total foreign membership	161
Total membership in United States	2,577
* Present address unknown	2
Total membership	2,740

SUMMARY OF MEMBERSHIP BY GRADES.

Honorary members	18
Members	1,856
Associates	221
Junior members	645
Total membership	2,740
† Life members	108

* These are Lawrence V. Melville and Frank Pettit, both Junior Members, and if any member knows their present addresses he will confer a favor by so advising the Secretary.

† These Life Members are included in the total membership above, in the class to which they belong.

INDEX

TO THE

CONSTITUTION, BY-LAWS AND RULES.

AMERICAN SOCIETY MECHANICAL ENGINEERS.

AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

CONSTITUTION.

NAME, OBJECT AND GOVERNMENT.

C 1. The title of this Society is "The American Society of Mechanical Engineers."

C 2. The object of the Society is to promote the Arts and Sciences connected with Engineering and Mechanical Construction. The principal means for this purpose shall be the holding of meetings for the reading and discussion of professional papers, and for social intercourse; the publication and distribution of its papers and discussions; and the maintenance of an Engineering Library.

C 3. The Society shall be governed by this Constitution, and by By-Laws and Rules in harmony therewith.

C 4. The Society was organized as a Corporation under the laws of the State of New York, April 7, 1880. Its offices shall be located in the City of New York.

MEMBERSHIP.

C 5. Persons connected with the Arts and Sciences relating to Engineering or Mechanical Construction may be eligible for admission into the Society.

C 6. The membership of the Society shall consist of Honorary Members, Members, Associates and Juniors. Honorary Members, Members and Associates are entitled to vote and to hold office. Juniors shall not be entitled to vote nor to be officers of the Society, but shall be entitled to the other privileges of membership.

C 7. Honorary Members, Members and Associates are en-

titled to vote on all questions before any meeting of the Society, in person or by proxy, given to a voting member. A proxy shall not be valid for a greater time than six months.

C 8. Honorary Members shall be persons of acknowledged professional eminence, and their number shall not exceed twenty-five at any time.

C 9. A Member must have been so connected with Engineering as to be competent, as a designer or as a constructor, to take responsible charge of work in his branch of Engineering, or he must have served as a teacher of Engineering for more than five years. A Member shall be thirty years of age or over.

C 10. An Associate must either have the other qualifications of a Member or be so connected with Engineering as to be competent to take charge of engineering work, or to co-operate with Engineers. An Associate shall be twenty-six years of age or over.

C 11. A Junior must have had such engineering experience as will enable him to fill a responsible subordinate position in engineering work, or he must be a graduate of an engineering school. A Junior shall be twenty-one years of age or over.

C 12. The rights and privileges of every Honorary Member, Member, Associate and Junior shall be personal to himself, and shall not be transferable or transmissible by his own act or by operation of law.

ADMISSION.

C 13. Honorary Members shall be nominated by at least ten members of the Society. The grounds upon which the nomination is made, shall be presented to the Council in writing.

C 14. All applications for membership to the grades of Member, Associate or Junior shall be presented to the Council, which shall consider and act upon each application, assigning each approved applicant to the grade of membership to which, in the judgment of the Council, his qualifications entitle him. The name of each candidate thus approved by the Council, shall, unless objection is made by the applicant, be submitted to the voting membership for election, by means of a letter-ballot.

C 15. Associates or Juniors desiring to change their grade of membership shall make application to the Council in the same manner as is required in the case of a new applicant.

C 16. Election to membership shall be by a sealed letter-ballot as the By-Laws shall provide. Adverse votes to the number of two per cent. of the votes cast shall be required to defeat the election of an applicant for any grade of membership. The Council, may in its discretion, order a second ballot upon a defeated applicant, in which case adverse votes to the number of four per cent. of the votes cast, shall be required to defeat the election.

C 17. The election of Honorary Members shall be by a vote of the Council taken by letter-ballot, as provided in the By-Laws. One dissenting vote shall defeat such election.

C 18. Each person elected, excepting Honorary Members, shall subscribe to this Constitution, and shall pay the initiation fee before he can be entitled to the rights and privileges of membership. If such person does not comply with this requirement within six months after notice of his election, he will be deemed to have declined election. The Council may, thereupon, declare his election void.

INITIATION FEES AND DUES.

C 19. The initiation fee for membership in each grade shall be as follows:

For Member	Twenty-five Dollars,
For Associate	Twenty-five Dollars,
For Junior	Fifteen Dollars.

C 20. A Junior, on promotion to any other grade of membership, shall pay an additional fee of Ten Dollars.

C 21. The annual dues for membership in each grade shall be as follows:

For Member	Fifteen Dollars,
For Associate	Fifteen Dollars,
For Junior	Ten Dollars for the first six years of his membership and thereafter the same as for an Associate.

C 22. The Council may in its discretion, permit any Member or Associate to become a Life Member in the same grade, by the payment at one time of an amount sufficient to purchase from the Equitable Life Assurance Society of New York, an annuity on the life of a person of the age of the applicant equal to the

annual dues in his grade. Such Life Member shall not be liable thereafter for annual dues.

C 23. The Council shall have the power, by letter-ballot, to admit to Life Membership, without the payment of a life membership fee, any person who, for a long term of years, has been a Member or an Associate when, for special reasons, such procedure would, in its judgment, promote the best interests of the Society, provided that notice of such proposed action shall have been given at a previous meeting of the Council. One dissenting vote shall defeat such admission.

SUSPENSIONS AND EXPULSIONS.

C 24. Any Member, Associate or Junior who shall leave his annual dues unpaid for one year, shall not receive the volume of *Transactions* until such arrears are paid. Any Member, Associate or Junior who shall leave his dues unpaid for two years, shall, in the discretion of the Council, have his name stricken from the roll of membership, and shall cease to have any further rights as such.

C 25. The Council may refuse to receive the dues of any member of any grade, who shall have been adjudged by the Council to have violated the Constitution or By-Laws of the Society, or who, in the opinion of the Council by a two-thirds vote, shall have been guilty of conduct rendering him unfit to continue in its membership; and the Council may expel such person and remove his name from the list of members.

THE COUNCIL.

C 26. The affairs of the Society shall be managed by a Board of Directors chosen from among its Members and Associates, which shall be styled "The Council." The Council shall consist of the President of the Society, who shall be the presiding officer, six Vice-Presidents, nine Managers, the Treasurer and five Past Presidents. Five members of the Council shall constitute a quorum for the transaction of business. The Secretary may take part in the deliberations of the Council, but shall not have a vote therein. The Chairman of the Finance Committee shall attend the meetings of the Council and take part in the discussion of financial questions but shall not have a vote.

C 27. The five surviving Past Presidents who last held the office shall be members of the Council with all the rights, privileges and duties of the other members of the Council.

C 28. The Council thus constituted shall be the legal Trustee of the Society. All gifts or bequests not designated for a specific purpose shall be invested by the Council, and only the income therefrom may be used for current expenses.

C 29. Should a vacancy occur in the Council, or in any elective office except the presidency, through death, resignation or other cause, the Council may elect a Member or Associate to fill the vacancy until the next annual election.

C 30. The Council shall regulate its own proceedings, and may by resolution delegate specific powers to an Executive Committee or to any one or more members of the Council. No act of the Executive Committee or of a delegate shall be binding until it has been approved by a resolution of the Council.

C 31. The Council shall present at the Annual Meeting of the Society a report verified by the President or Treasurer or by a majority of the members of the Council, showing the whole amount of real and personal property owned by the Society, where located, and where and how invested, and the amount and nature of the property acquired during the year immediately preceding the date of the report, and the manner of the acquisition; the amount applied, appropriated or expended during the year immediately preceding such date, and the purposes, objects or persons to or for which such applications, appropriations or expenditures have been made; also the names and places of residence of the persons who have been admitted to membership in the Society during the last year, which report shall be filed with the records of the Society, and an abstract thereof shall be entered in the minutes of the proceedings of the Annual Meeting.

C 32. An act of the Council, which shall have received the expressed or the implied sanction of the membership at the next subsequent meeting of the Society, shall be deemed to be the act of the Society, and shall not afterwards be impeached by any member.

C 33. The Council may, by a two-thirds vote of the members present, declare any elective office vacant, on the failure of its incumbent for one year, from inability or otherwise, to attend the Council meetings, or to perform the duties of his office, and shall thereupon appoint a Member or Associate to fill the vacancy

until the next Annual Meeting. The said appointment shall not render the appointee ineligible to election to any office.

OFFICERS.

C 34. At each Annual Meeting there shall be elected from among the Members and Associates:

A President to hold office for one year.

Three Vice-Presidents, each to hold office for two years.

Three Managers, each to hold office for three years.

A Treasurer to hold office for one year.

C 35. The election of officers shall be by sealed letter-ballot, as the By-Laws shall provide.

C 36. The term of all elective officers shall begin on the adjournment of the Annual Meeting of the Society. Officers shall continue in their respective offices until their successors have been elected and have accepted their offices.

C 37. A President, Vice-President or Manager shall not be eligible for immediate re-election to the same office at the expiration of the term for which he was elected.

C 38. The Council, at its first meeting after the Annual Meeting of the Society, shall appoint a person of the grade of Member to serve as Secretary of the Society for one year, subject to removal for cause by an affirmative vote of fifteen members of the Council, at any time after one month's written notice has been given him to show cause why he should not be removed, and he has been heard in his own defense, if he so desires. The Secretary shall receive a salary which shall be fixed by the Council at the time of his appointment.

C 39. The President, Secretary and Treasurer shall perform the duties legally or customarily attaching to their respective offices under the Laws of the State of New York, and such other duties as may be required of them by the Council.

C 40. A vacancy in the office of President shall be filled by the Vice-President, who is senior by age.

MEETINGS.

C 41. The Society shall hold two meetings in each year. The Annual Meeting shall begin in New York City on the first Tuesday in December, and a Semi-Annual Meeting shall be held at

such time and place as the Council may appoint. Fifty Members and Associates shall constitute a quorum for the transaction of business.

C 42. Special meetings of the Society may be called at any time at the discretion of the Council, or shall be called by the President upon the written request of fifty members entitled to vote, the notices for such meetings to state the business for which such meeting is called, and no other business shall be entertained or transacted at that meeting.

C 43. Any appropriation recommended by the Society at a meeting shall not take effect until it has been approved by the Council.

C 44. Every question which shall come before a meeting of the Society or of the Council or a Committee, shall be decided by a majority of the votes cast, unless otherwise provided in this Constitution or the By-Laws, or the Laws of the State of New York. The Council may order the submission of any question to the membership for discussion by letter-ballot. Any meeting of the Society at which a quorum is present, may order the submission of any question to the membership for discussion by letter-ballot.

STANDING COMMITTEES.

C 45. The Standing Committees of the Society to be appointed by the President shall be:

Finance Committee,
Committee on Meetings,
Publication Committee,
Membership Committee,
Library Committee,
House Committee.

C 46. There shall be a John Fritz Medal Committee of three members appointed as provided in the By-Laws.

C 47. The Annual Committees shall be:

An Executive Committee, appointed by the Council.
A Nominating Committee, appointed by the President.
Tellers as required by the By-Laws, appointed by the President.

C 48. Special Nominating Committee:

Twenty or more members entitled to vote may constitute

themselves a Special Nominating Committee, with the same powers as the Annual Nominating Committee.

C 49. Professional Committees:

The Council shall have power to appoint, upon a recommendation of the Society at a general meeting, or upon its own initiative, such Professional Committees as it may deem desirable, to investigate, consider and report upon subjects of engineering interest. Reports of such committees may be accepted by the Society and printed in the *Transactions*, but shall not be approved or adopted as the action of the Society. Any proposed expenses of such committees must be authorized by the Council before they are incurred.

C 50. Each Committee shall perform the duties required of it in the By-Laws, or assigned to it by the Council. The Secretary of the Society shall be the Secretary of each of the Standing Committees.

C 51. The Council may at any time, in its own discretion, remove any or all members of any Committee, except a Nominating Committee; and the vacancy, arising from this or from any other cause, shall be filled by appointment by the President, except a vacancy in the Executive Committee, which shall be filled by the Council.

SECTIONS OF THE SOCIETY.

C 52. The Council may, in its discretion, authorize the organization of sections or groups of any or all grades of membership, for professional or scientific purposes which are in harmony with the Constitution and By-Laws of this Society. Such sections or groups may, in the discretion of the Council, be geographical or professional, and shall have such powers, and act under such rules and regulations as the Council may from time to time prescribe.

TRANSACTIONS.

C 53. The official record of technical papers and discussion, shall be known as the *Transactions* of the Society, and shall be published under the direction of the Council. There may be included therein, the annual report of the Council, reports of Committees, and business records of the Society.

C 54. The Society shall claim no exclusive copyright to any papers read at its meetings, or any reports or discussions thereon, except in the matter of their official publication under the Society's imprint as its *Transactions*. The policy of the Society shall be to give the professional and scientific papers read before it the widest circulation possible, with the view of making the work of the Society known, encouraging Engineering progress and extending the professional reputation of its members.

C 55. The Society shall not be responsible for statements or opinions advanced in papers or in discussions at its meetings. Matters relating to politics or purely to trade shall not be discussed at a meeting of the Society, nor be included in the *Transactions*.

C 56. The Society shall not approve or adopt any standard or formula, or approve any engineering or commercial enterprise. It shall not allow its imprint or name to be used in any commercial work or business.

AMENDMENTS TO THE CONSTITUTION.

C 57. At any semi-annual meeting of the Society any member may propose in writing an amendment to this Constitution. Such proposed amendment shall not be voted on at that meeting, but shall be open to discussion and to such modification as may be accepted by the proposer. The proposed amendment shall be mailed in printed form by the Secretary to each member of the Society entitled to vote, at least sixty days previous to the next annual meeting, accompanied by comment by the Council, if it so elects. At that annual meeting such proposed amendment shall be presented for discussion and final amendment, and shall subsequently be submitted to all members entitled to vote, provided that twenty votes are cast in favor of such submission. The final vote on adoption shall be by sealed letter-ballot, closing at twelve o'clock noon on the first Monday of March following.

C 58. The letter-ballot, accompanied by the text of the proposed amendment, shall be mailed by the Secretary to each member of the Society entitled to vote at least thirty days previous to the closure of the voting. The ballots shall be voted, canvassed and announced as provided in the By-Laws. The adoption of the amendment shall be decided by a majority of the

votes cast. An amendment shall take effect on the announcement of its adoption by the Presiding Officer of the semi-annual meeting next following the closure of the vote.

AMENDMENTS TO BY-LAWS AND RULES.

C 59. For the further ordering of the affairs of the Society, the Council may, by a two-third vote of its members present, amend the By-Laws in harmony with this Constitution, provided that a written notice of such proposed amendment shall have been given at the previous regular meeting of the Council; and provided further that the Secretary shall have mailed to each member of the Council a copy of such proposed amendment, at least thirty days in advance of the meeting of the Council at which action is to be taken. The amendment shall take effect immediately on its passage by the Council. The Secretary shall at once mail a copy of such amendment to the members of all grades.

C 60. The Council may, by a majority vote of the members present at any meeting, establish, amend or annul Rules for the conduct of the business affairs of the Society; for the ordering and conduct of its professional or business meetings; and for guidance of its committees in their work and reports; provided that such Rules are in harmony with the Constitution and By-Laws of the Society.

CONSTITUTION GOES INTO EFFECT.

C 61. This Constitution shall supersede all previous Rules of the Society, and shall go into effect on the announcement by the Presiding Officer of its adoption.

BY-LAWS.

CANDIDATES FOR MEMBERSHIP.

B 1. A candidate for admission to the Society as a Member or as an Associate must make application on a form approved by the Council, upon which he shall write a statement giving a complete account of his qualifications and engineering experience, and an agreement that he will, if elected, conform to the Con-

stitution, By-Laws and Rules of the Society. He must refer to at least five Members or Associates to whom he is personally known.

B 2. Applications for membership from Engineers who are not resident in the United States or Canada, and who may be so situated as not to be personally known to five Members of the Society, as required in the foregoing paragraph, may be recommended for ballot by five members of the Council, after sufficient evidence has been secured to show that in their opinion the applicant is worthy of admission to the grade which he seeks.

B 3. A candidate for admission to the Society as a Junior must make application in the same manner as provided for Members, except that he must refer to not less than three Members or Associates to whom he is personally known.

B 4. References shall not be required of candidates for Honorary Membership.

B 5. The references for each candidate for admission to the Society shall be requested to make a confidential communication to the Membership Committee, setting forth in detail such information, personally known to referee, as shall enable the Council to arrive at a proper estimate of the eligibility of the candidate for admission to the Society.

ELECTION OF MEMBERS.

B 6. The Secretary shall mail to each member entitled to vote, at least thirty days in advance of each annual or semi-annual meeting, a ballot stating the names and the respective grades of the candidates for membership in the Society which have been approved by the Council, and the time of the closure of voting. The voter shall prepare his ballot by crossing out the names of candidates rejected by him, and shall enclose said ballot in a sealed blank ballot envelope which he shall then enclose in a second sealed outer envelope on which he shall, for identification, write his name in ink. The ballot thus prepared and enclosed shall be mailed or delivered unopened to the Tellers of Election. The Secretary shall certify to the competency, and the signature of all voters. On the closure of voting, the Tellers of Election shall first open and destroy the outer envelopes, and shall then canvass the ballots, and certify the result to the meeting of the Society.

B 7. The Tellers shall not receive any ballot after the stated time of the closure of voting. A ballot without the endorsement of the voter, written in ink on the outer envelope, is defective, and shall be rejected by the Tellers of Election.

B 8. The names of those persons elected to membership, with their respective grades, shall be embodied in a written report, signed by the Tellers, and presented to the next meeting of the Society. The President shall then declare them duly elected to membership in the Society. The Tellers may, through the Secretary, in advance of any meeting advise each candidate of the result of the canvass of the votes in his case. The names of applicants who are not elected shall neither be announced nor recorded in the *Transactions*.

B 9. The endorsers of an applicant who has not been elected, may, with his consent, present to the Council a written request for a re-submission of his name to ballot. The Council may, in its discretion, by a three-fourths vote of the members present, order the name of the applicant placed on the next ballot for members.

B 10. Election to Honorary Membership shall be by letter-ballot of the Council. A notice of such proposed election shall be mailed by the Secretary to each member of the Council at least sixty days in advance of the date set for the closure of such election.

B 11. Each person elected to membership, except an Honorary Member, must subscribe to the Constitution, By-Laws and Rules of the Society, and pay the initiation fee before he can receive a certificate of membership in the Society.

ELECTION OF OFFICERS.

B 12. The Secretary shall mail to each member entitled to vote, at least thirty days before the Annual Meeting, the names of the candidates for office proposed for election by the Nominating Committee.

B 13. The names of the candidates proposed by the Nominating Committee or Committees, and the respective offices for which they are candidates, shall be printed in separate lists on the same ballot sheet, each list of candidates to be printed under the names of the members of the particular committee which proposed it.

B 14. The name of any candidate on the ballot may be erased, and the name of any person qualified to hold the office written in its stead. The voter shall make a cross with a pen or pencil before the name of each candidate for office for whom he wishes to vote. The ballot thus prepared must be voted and canvassed in the same manner as for the election of members.

B 15. At the first session of the Annual Meeting, the Tellers of Election of Officers shall canvass the votes cast for the officers of the Society in the manner prescribed for the election of members, and immediately report the result of the canvass to the meeting. The President shall then announce the candidates having the greatest number of votes for their respective offices, and declare them elected for the ensuing year.

B 16. In case of a tie in the vote for any officer, the President or, in his absence, the Presiding Officer shall cast the deciding vote.

B 17. A ballot which contains more names marked by a cross on it than there are officers to be elected, is thereby defective, and shall be rejected by the Tellers.

FEES AND DUES.

B 18. The initiation fee and annual dues of the first year shall be due and payable on notice of election to membership, and upon that payment the member will be entitled to the *Transactions* for the year. Thereafter the annual dues shall be due and payable on the first day of October in each year.

B 19. A member in arrears for one year shall not be entitled to vote until such arrears have been paid. Should the right to vote be questioned, the books of the Society shall be conclusive evidence.

B 20. The Secretary shall present to the Council the name of any Member, Associate or Junior in arrears for more than one year, and such member shall not receive the *Transactions* until such arrears are fully paid. A person dropped from the rolls for non-payment of dues may, in the discretion of the Council, be restored to the privileges of membership, upon payment of all arrears.

FINANCIAL ADMINISTRATION.

B 21. The Council at its first meeting in each fiscal year, shall consider the recommendations of the Finance Committee

concerning the expenditure necessary for the work of the Society during that year. The apportioning of the work of the Society among the various Standing and other Committees shall be on a basis approved by the Council and in harmony with the Constitution and By-Laws. The appropriations approved by the Council, or so much thereof as may be required for the work of the Society, shall be expended by the various Committees of the Society, and all bills against the Society for such expenditure shall be certified by the Committee making the expenditure and shall then be sent to the Finance Committee for audit. Money shall not be paid out by any officer or employee of the Society except upon bills duly audited by the Finance Committee, or by resolution of the Council.

COMMITTEES.

B 22. The President within one month after the Annual Meeting shall fill all vacancies in the Standing Committees by appointment from the membership of the Society.

Each of the Standing and the Annual Committees, shall, at their first meeting after the Annual Meeting, elect a Chairman to serve for one year. The President shall appoint the Chairman of each Professional Committee. A member of a Standing Committee whose term of office has expired, shall continue to serve until his successor shall have been appointed.

FINANCE COMMITTEE.

B 23. The Finance Committee shall consist of five Members or Associates. The term of office of one member of the Committee shall expire at the end of each Annual Meeting. This Committee shall, in the discretion of the Council, have a supervision of the financial affairs of the Society, including the books of account. The Committee may cause the accounts of the Society to be audited and approved annually by a chartered or other competent public accountant. The Committee shall hold monthly meetings for the audit of bills and such other business as shall come before it and shall deliver to the Secretary for presentation to the Council at the end of each fiscal year, a report of the financial condition of the Society for the past year, and also shall present therewith a detailed estimate of the prob-

able income and expenditure of the Society for the following twelve months. It shall make recommendations to the Council as to investments, and, when called upon by the Council, advise upon financial questions.

COMMITTEE ON MEETINGS.

B 24. The Committee on Meetings shall consist of five persons who may be members of any grade. The term of office of one member of the Committee shall expire at the end of each Annual Meeting. It shall be the duty of the Committee to procure professional papers, to pass upon their suitability for presentation, and to suggest topical subjects for discussion at the meetings. The Committee may refer any paper presented to the Society to a person or persons, especially qualified by theoretical knowledge or practical experience, for their suggestions or opinions as to the suitability of the paper for presentation. Papers from non-members shall not be accepted except by unanimous vote of the Committee.

The Committee shall arrange the programme of each meeting of the Society, and shall have general charge of the entertainments to be provided for the members and guests at each meeting. It shall prohibit the distribution or exhibition at the headquarters or at the meeting places of the Society of all advertising circulars, pamphlets or samples of commercial apparatus or machinery. At the end of each fiscal year, the Committee shall deliver to the Secretary for presentation to the Council, a detailed report of its work.

PUBLICATION COMMITTEE.

B 25. The Publication Committee shall consist of five Members or Associates. The term of office of one member shall expire at the end of each Annual Meeting. The Committee shall review all papers and discussions which have been presented at the meetings, and shall decide what papers or discussions, or parts of the same, shall be printed in the *Transactions* of the Society. The Committee will be expected to publish all such data as will be of assistance to engineers or investigators in their work. At the end of each fiscal year, the Committee shall deliver to the Secretary for presentation to the Council, a detailed report of its work.

MEMBERSHIP COMMITTEE.

B 26. The Membership Committee shall consist of five Members or Associates. The term of office of one member of the Committee shall expire at the end of each Annual Meeting. It shall be the duty of this Committee:

To meet monthly to receive and scrutinize all applications for membership to the Society.

To send to each voting member the name, qualifications, engineering experience and references of each applicant, together with extracts from the Constitution and By-Laws relating to membership.

To seek further information as to the qualifications of an applicant, whose evidence of eligibility is not clear to the Committee.

To report to each session of the Council the names of all applicants under consideration together with the action of the Committee on each.

The Committee shall at once destroy all correspondence in relation to each applicant when his name has been placed on the ballot by order of the Council, or upon the withdrawal of the application.

LIBRARY COMMITTEE.

B 27. The Library Committee shall consist of five Members, Associates or Juniors. The term of office of one member of the Committee shall expire at the end of each Annual Meeting. It shall be the duty of the Library Committee to take charge of the Library of the Society, the historical relics, the paintings and objects of art, and to recommend to the Council suitable regulations for their care and use. At the end of each fiscal year, the Committee shall deliver to the Secretary, a detailed report of its work.

HOUSE COMMITTEE.

B 28. The House Committee shall consist of five Members, Associates or Juniors. The term of office of one member of the Committee shall expire at the end of each Annual Meeting. It shall be the duty of the House Committee to have the care, management and maintenance of the house of the Society and its furnishings. They may make rules for the care and the use

of the Society House, subject to the approval of the Council. At the end of each fiscal year, the Committee shall deliver to the Secretary a detailed report of its work.

EXECUTIVE COMMITTEE.

B 29. The Council shall appoint from its members an Executive Committee to act for the Council during the interval between its sessions. The Committee shall make a report of its acts to each session of the Council for approval. The Secretary may take part in the deliberations of the Executive Committee, but shall not have a vote therein.

NOMINATING COMMITTEES.

B 30. A Nominating Committee of five Members, not members of the Council, shall be appointed by the President within three months after he assumes office. It shall be the duty of this Committee to send to the Secretary on or before October first the names of consenting nominees for the elective offices next falling vacant under the Constitution. Upon the request of any Member or Associate, the Secretary shall furnish to the applicant the names of such nominees.

B 31. A special Nominating Committee if organized, shall, on or before October twentieth, present to the Secretary the names of the candidates nominated by it for the elective offices next falling vacant under the Constitution, together with the written consent of each.

JOHN FRITZ MEDAL COMMITTEE.

B 32. The John Fritz Medal Committee shall consist of three persons of the grade of Member, to be appointed by the Council. The term of office of one member of this Committee shall expire at the end of each annual meeting. The duty of this Committee shall be to represent the Society in the Board of Trustees of the John Fritz Medal Fund Corporation.

REPRESENTATIVE DELEGATES.

B 33. The Council may in its discretion appoint a member or members of the Society or other person or persons to repre-

sent it at meetings of Societies of kindred aim or at public functions. Such delegates shall be designated as "Honorary Vice-Presidents," and their duties shall terminate with the occasion for which they were appointed.

TELLERS.

B 34. The Presiding Officer shall, at the first session of the Annual Meeting, appoint three Tellers of Election of officers, whose duties shall be to canvass the votes cast, and report the result to the meeting. Their term of office shall terminate when their report of the canvass is presented to the meeting.

B 35. The President within one month after assuming office shall appoint three Tellers of Election of members to serve for one year, whose duties shall be to canvass the votes cast for members during the year, and to certify the same to the President. They shall notify candidates through the Secretary of the result of such election.

B 36. The President shall appoint three Tellers to canvass any letter-ballots which shall be ordered by the Council or by the Society.

MEETINGS.

B 37. The meetings of the Society shall continue from day to day as the meeting may decide. The business session of the Annual Meeting shall be held on Wednesday following the first Tuesday of December. The professional sessions for the reading of papers shall be held at such times and places as the meeting may appoint. Notices of all meetings of the Society shall be mailed by the Secretary to members of all grades not less than thirty days before the date of such meeting.

SECRETARY.

B 38. The Secretary of the Society shall be the Secretary to the Council and also to each of the Standing Committees.

The Secretary shall, under the supervision of the Finance Committee, have charge of the Books of Account of the Society.

He shall make and collect all bills against members or others.

He shall have charge of all bills against the Society, shall

keep an account of the same, and shall present them in proper form to the Finance Committee for audit.

All funds received by any person for the Society, shall be delivered to the Secretary. He shall immediately enter them in the Books of Account, and shall immediately deposit such funds as he receives, to the credit of the Society, in a Bank to be designated by the Council.

TREASURER.

B 39. The Treasurer shall make payments only on the audit of the Finance Committee, or upon the direction of the Council, by resolution of that body. He shall furnish a bond for the faithful performance of his duties to such amount as the Council may require, such bond to be procured from an incorporated Guarantee Company, at the expense of the Society.

TITLES, EMBLEMS, CERTIFCATE.

B 40. Each Member and Associate shall, subject to such rules as the Council may establish, be entitled on request, to a certificate of membership, signed by the President and Secretary of the Society. Every such certificate shall remain the property of the Society, and shall be returned to it on demand of the Council.

B 41. Each proxy authorizing a person to vote for an absent member, shall be signed by such absent member, with an attesting witness, and be submitted to the Secretary for verification of the member's right to vote at the meeting at which the right is to be exercised.

B 42. The emblem of each grade of membership approved by the Council shall be worn by those only who belong to that grade. The official stationary shall be used only by Officers and Committees of the Society.

B 43. The abbreviation of the titles of the various grades of membership approved by the Society are as follows:

For Honorary Members, . .	Hon. Mem. Am. Soc. M. E.
For Members, ...	Mem. Am. Soc. M. E.
For Associates, ...	Assoc. Am. Soc. M. E.
For Juniors, ...	Jun. Am. Soc. M. E.

RULES.

R 1. The Secretary's office shall be open on business days from 9 A.M. to 5.30 P.M. During the Annual Meeting, the office shall be open from 9 A.M. to 10 P.M. A register shall be kept for each regular meeting, to record the attendance of members and guests.

R 2. The Secretary shall provide a numbered badge or pin for each member or guest attending the regular meetings, the number on the badges to correspond with the member's of guest's number on the register.

R 3. The Secretary shall at each regular meeting of the Society distribute at the headquarters a printed list of the names registered at the meeting.

R 4. Copies of papers to be read and discussed at any meeting shall be sent to each member thirty days in advance of that meeting. A paper received too late for such distribution shall only be accepted for presentation at that meeting by unanimous consent of the Committee on Meetings. A blank shall accompany the papers by which a member may signify his intention to discuss any of the papers, and priority in debate shall be given in the order of the receipt by the Secretary of such notification.

R 5. At professional sessions, each paper shall be read by abstract only, ten minutes being allowed to the author for the presentation, unless otherwise ordered by the meeting.

R 6. A member who has given notice of his intention to discuss a paper, and shall have reduced his discussion to writing, shall be entitled to ten minutes for its presentation.

R 7. Each speaker shall be limited to five minutes in the oral discussion of a paper, unless the time should be extended by unanimous consent. A member who has once had the floor cannot claim it again until all the others have been heard who desire to speak on that paper. Authors may have five minutes to close the discussion on the paper.

R 8. Members unable to attend the meeting may send a discussion of any paper in writing, to be presented by the Secretary.

R 9. The Committee on Meetings shall deliver to the Secretary such papers as they recommend for presentation to the professional meetings of the Society.

R 10. The Secretary shall have sole possession of papers and illustrations between the time of their approval by the Committee on Meetings, and their presentation to the professional session of the Society.

R 11. After the presentation and discussion of a paper, a copy of both shall be sent to the author, and, so far as possible, a copy of the reported discussion shall be sent to each member who presented it, with the request that he correct errors or omissions, and return the same promptly to the Secretary.

R 12. Members may order reprints of papers at a price sufficient to cover the cost to the Society, provided that said copies are not for sale.

R 13. The Secretary may furnish to the author twenty copies of his paper without charge. He may also furnish to the technical press such papers in advance of the meeting as they may wish to publish after presentation to the meeting of Society.

R. 14. The entertainments to be provided for the members and guests at any meeting of this Society in any city shall be in charge of a Local Committee, subject, however, to the general approval of the Committee on Meetings.

R 15. A member may invite a non-member to the professional sessions of the meeting, but the guest shall not take part in the proceedings without an invitation from the Presiding Officer. Invitations to guests of members for the entertainments provided for the Society shall be in the discretion of the Local Committee.

R 16. The Society House shall be open at all hours for access to members. The Library shall be open on all week days between the hours of 10 o'clock A.M. and 10 o'clock P.M. It shall be conducted as a Free Public Reference Library of Engineering and the Allied Arts and Sciences.

R 17. Juniors who were elected to membership in the Society six years or more previous to the adoption of this Constitution, shall pay the same dues as an Associate, beginning with the fiscal year which opens after such adoption. Juniors, who have been elected less than six years before that date, shall pay the dues of an Associate on the expiration of six years after their election.

CONTENTS OF VOLUME XXV.

NEW YORK (48TH) MEETING.

CHICAGO, ILL. (49TH) MEETING.

LIST OF ILLUSTRATIONS.

PAPERS

OF THE

NEW YORK MEETING

(XLVIIIth)

OF THE

AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

DECEMBER 1st TO 4th, 1903.

BEING ALSO THE TWENTY-FOURTH ANNUAL MEETING OF THE SOCIETY.

No. 1007.

PROCEEDINGS

OF THE

NEW YORK MEETING

(XLVIIIth)

OF THE

AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

December 1st to 4th, 1903.

Opening Session. Tuesday, December 1st, 1903.

The twenty-fourth annual meeting of the Society, which was also its forty-eighth convention, was held in New York City, at the house of the Society and its Library, No. 12 West Thirty-first Street, during the days December 1st to 4th, 1903.

The opening session was called to order by the President of the Society, Mr. James M. Dodge of Philadelphia, at nine o'clock on Tuesday evening.

It became apparent at this first session that the meeting was to be one of phenomenal size in the matter of members in attendance, and the audience crowded the auditorium to listen to the address of the President.

After a few words of salutation from the Chair, Messrs. Lane, Tompkins and Kern Dodge were appointed tellers under the provisions of the Rules, to count the Officers' Ballot, to be presented at the next succeeding session, and Messrs. La Forge and Louer were appointed tellers to count the letter ballots on the

Amendments to the new Rules, which were to constitute the new Constitution, By-Laws and Rules of the Society.

The President then delivered his address, entitled "The Money Value of Technical Training," which appears as one of the papers of this meeting.

After announcements by the Secretary concerning the meeting, a recess was taken to partake of light refreshments served in the collation room and for an informal reunion of members.

SECOND SESSION. WEDNESDAY MORNING, DECEMBER 2ND.

The second or business session of the annual meeting was called to order at ten o'clock in the Hall of the Mendelssohn Union, 113 West Fortieth Street. This step was made necessary by the limited accommodation in the auditorium of the Society, which compelled a choice of a larger meeting room.

The headquarters for registration and other executive business was retained at the house of the Society, 12 West Thirty-first Street.

The registration of this session was made noteworthy by the first effort to combine the system in use at previous meetings, with the wearing of an inconspicuous tag under the lapel button and number, which carried the name of the member so that it could be read at short distances. It was believed that by this procedure the comparative awkwardness of hunting up a man's name on the printed register lists would be removed, and the social approach of members to each other would be stimulated. The smooth working of the plan in its tenative form justified the experiment, and until further notice it will be carried out. Up to the adjournment of the meeting on Friday morning there were 823 names registered, of which 538 were members. This is the largest enrollment of members in the history of the Society.

The first matter of business of the session was the presentation to the meeting of the Annual Report of the Council and Standing Committees of the Society.

These reports had been printed and distributed to all members, in advance of the meeting, for their information; they were read by the Secretary, by title, and are presented herewith for record.

Some minor changes in distribution of accounts, in the Report of the accountant, were presented verbally, but which did not alter the totals, merely detail of allotment of expenses to certain accounts.

The Reports were as follows:

ANNUAL REPORT OF THE COUNCIL.

Fiscal year, 1902–1903.

The Council presents herewith, as required by the Rules of the Society, a report of business which has passed under its hand during the year which closes with the annual meeting in December, 1903.

The most important business of the year has been the announcement of the munificent purpose of Mr. Andrew Carnegie, a member of the Society, to give the sum necessary to make adequate provision for the accommodation of the four national societies of engineers, in an appropriate building, and for the Engineers' Club in another. Mr. Carnegie expressed his willingness to make the amount of his gift exceed a minimum of one million dollars, if that sum should be found inadequate to give the accommodation required for the present and the future needs of the organizations which he named. The Council was convened in special session on the afternoon of Thursday, May 7th, to consider the simple proposition of Mr. Carnegie's letter, at which eleven members of the Statutory Council, and seven past presidents of the Society were present. The Council has made full report to the membership by circular of the resolutions which were passed at that and a subsequent meeting, concerning the Carnegie gift, which have been made matters of official record in the Transactions of the Forty-seventh Meeting at Saratoga. The Council has also issued full bulletins to the members, which were also made part of that record.

The four constituent societies named by Mr. Carnegie have appointed representatives, and from these representatives an Executive Committee has been formed, which since the adjournment of the Saratoga Meeting has been formulating the details of the arrangement of the building, and another sub-committee has been considering the proper method for the management and control of such a joint undertaking.

The Report of these committees will be made public in the near future.

The Council has convened for routine business at the necessary intervals during the year.

It took favorable action at its first meeting concerning the

issue of a letter ballot whereby the individual members of the Society might express their opinion as to the effect of the adoption of legislation making the metric system compulsory on citizens of the United States. The result of such ballot, with the expression of such opinion, was reported at the Saratoga Meeting.

The Council accepted on the first of February, the resignation of Mr. Arthur L. Rice, who had been acting as Assistant to the Secretary, and confirmed the engagement of Mr. Louis A. Gillet, under a different financial arrangement.

The Council has received communications during the winter from members in different cities, concerning the probable attitude which it would take with respect to the formation of local chapters of the Society. The Council has in every case directed that until the Society had taken official action upon that provision in its proposed Constitution, looking towards the formation of sections of the Society, that it was premature to open discussion on these details.

Mr. C. J. H. Woodbury of Boston had been asked to represent the Society at a conference for the revision of the National Standard Electrical Rules, and as such representative has furnished a report of the action of the conference.

An arrangement has been made with the Engineers' Club of Philadelphia whereby the privileges of the house of that Club, in Philadelphia, may be enjoyed by members of the American Society of Mechanical Engineers, and similarly that members of the Club may have the privilege of use of the house of the A. S. M. E. in New York City, on the presentation of the respective cards of introduction issued by the two organizations to their respective members.

The Council has expressed the interest of the Society in coöperating with the American Reception Committee of the Iron and Steel Institute of Great Britain in their undertaking to provide for a meeting in 1904, of that organization, in the United States.

The Council has acted favorably upon a request that a provisional committee should be created to consider and report upon a tonnage basis for expressing the effectiveness of refrigerating machinery, which could be made generally acceptable as a standard. This Committee consists of Professor D. S. Jacobus, Chairman, Messrs. E. F. Miller, A. P. Trautwein, G. T. Voorhees, P. De Catesby Ball.

The Council has had under consideration the invitations which

have been received from those interested in the success of the Louisiana Purchase Exposition in St. Louis, which has urged that the Society shall select the City of St. Louis as a convention city, during the summer of 1904, while the exposition was in progress.

It was the sense of the Council that it would be more serviceable if the convention of that date should be held in a city within convenient access by rail to the exposition city rather than in St. Louis itself.

On communicating this opinion to the representative members in the City of Chicago, Ill., the outcome has been a most emphatic urging that Chicago should be fixed upon as the point for the spring convention in the exposition year. The Council has acted favorably on this invitation and the City of Chicago has been selected.

It has been further decided by the Council to avail of this opportunity to invite the Institution of Mechanical Engineers of Great Britain to hold an American convention, and that such convention be a joint session with the Society of Mechanical Engineers, at Chicago, with a view to having such guests of the Society as might come from Great Britain, within convenient access of the exposition city, upon the same journey which brought them to the meeting.

The invitation of the Council has been cordially accepted by the Institution of Mechanical Engineers of Great Britain, and the details of such joint meeting are in progress. The Institution of Civil Engineers of Great Britain was also invited at the same time, but an invitation to a similar joint meeting in September, and its acceptance by the Institution, made it impossible that our invitation to that society should be accepted in any official way for the month of May.

The Council directed that the practice should prevail for the present of having the Report of the Society's accountant audited each year by a firm of public accountants, such as the Audit Company of New York, or similar competent authority.

The Council has considered a request to undertake the responsibility of organizing the Section of Mechanical Engineering in the suggested International Congress of Engineering, in connection with the St. Louis Exposition.

It was the sense of the Council after discussion, that in the absence of a strong demand from the profession itself, for the hold-

ing of such a Congress, that it would not be advisable that this responsibility should be undertaken. The Council has decided to maintain a headquarters in the gallery of Machinery Hall of the exposition buildings for the convenience and use of members of the Society and its guests and the necessary appropriation has been made for the employment of a suitable person to maintain such headquarters and attend to the necessary clerical detail involved. It is proposed that such headquarters should be a centre for registration of members in attendance, and for the dissemination of information concerning the exposition to visiting members, but that it should not be maintained as an exhibit of the achievements of the profession.

The Council, on being advised of the sudden death of Professor Robert H. Thurston, the first President of the Society and serving for two terms, from 1880-1882, has directed the entry on its minutes of a Memorial Tribute, and that such tribute be made a matter of record and presentation at the general meeting of the Society.

The Council has passed votes of thanks to Miss Louisa Lee Schuyler and Mr. W. A. Gabriel, and others for gifts to the Society received during the current fiscal year.

The Council would present for record the list of members who have died during the current year as follows:

James Spiers, August 13, 1902; W. W. Lindsay, November 12th; Thos. J. Borden, November 22d; George Leach, November 27th; Geo. R. Fulton, December 4th; J. F. Pajeken, December 16th; P. F. Greenwood, December 22d; J. O. Nixon, January 3, 1903; A. Christensen, January 16th; David P. Jones, January 1, 1903; John Hulett, January 31st; Wm. Harkness, February 28th; Chas. M. Day, February; Victor Mackiewicz, February; Edward A. Darling, March 16th; John P. McGuire, April 17th; Irving M. Scott, April 28th; Elihu Dodds, June 10th; George Shaw, May 28th; Edward Graftstrom, June; George S. Morison, July 1st; Wm. Garrett, July 15th; E. H. Messer, August 12th; Wm. H. Stratton, August 13th; John Humphrey, August 24th; S. J. Geoghegan, September 7th; Pulaski Leeds, September 8, 1903; L. C. Crowell, September 16th; Wm. P. Canning, September 17, 1903; J. Q. Wright, October 16th; Robert H. Thurston, October 25, 1903; Y. Aisawa, October, 1903; Sir Fred'k Bramwell, December 1, 1903.

Pursuant to a desirable policy inaugurated a year ago, the

Council would call attention to the report of its Standing Committees covering the Society's work during the year.

The Reports of the Library and House Committee, and the Report of the Executive Committee, and of the Publication Committee, will be self-explanatory. With respect to the Report of the Finance Committee the Council would call attention to the following facts and deductions from that Report:

Items deduced from the accounts of fiscal year 1902-3 *showing the expense incurred per member:*

(1) Total members as per July, 1903, catalogue		2,573	
Deduct for members who have paid no dues:			
Life members	107		
Deaths and resignations	6		
Lapsed memberships	35		
Members who have not paid current year at September 30, 1903	101—	249	
Paying membership, 1902-3		2,324	
(2) Total income exclusive of 1 per cent. from dues carried to Library Development Fund, 90 per cent. from initiation fees, entire life membership receipts, carried to Reserve Fund, and entire Sinking and Fellowship Fund, subscriptions of Mechanical Engineers' Library Association		$38,662 03	
Income earned per paying member (computed)		16 63	
Income earned per paying member, dues only (computed)		14 20	
(3) Total expense incurred year October 1, 1902, to September 30, 1903, less cost operating house ($3,347.54), mortgage interest ($1,402 50), repairs and renewals ($644.32), depreciations house and furniture ($481.45)—$31,773 55:			
(4) Total expense incurred for publications, October 1, 1902, to September 30, 1903		14,956 74	
(5) Total expense incurred for salaries in Society's office same period		9,308 55	
(6) Total expense incurred for all other accounts except house		7,508 26—	31,773 55
(7) Total expense incurred for house, including interest on mortgage, repairs and renewals and depreciations		5,875 81	
Deduct income earned from rent of sleeping rooms and hall		2,002 25—	3,873 56
Net expense incurred for year			$35,647 11

(Gross expense, $37,649.36 less rental income $2,002.25, equals $35,647.11).

Expense incurred per paying member, October 1, 1902, to September 30, 1903:

(8)	For all purposes including house		$15 33
(9)	For house operation including interest and repairs		1 66
(10)	For all purposes exclusive of house		13 67
(11)	For publications, printers' work, engraving, binding and distribution	$6 43	
(12)	For salaries in Society's office	4 04	
(13)	For all other expenses except house	3 20—	13 67
(14)	For house operation exclusive of mortgage interest, repairs and renewals and depreciations		57
(15)	For house operation exclusive of mortgage interest, but including repairs, renewals and depreciations		1 06
(16)	For operating library		31
(17)	For postage, circulars, catalogues, and stationary and printing in Society's office		1 83
(18)	For meetings, and all other expense not otherwise allotted above		1 26

Comparative income earned with expense incurred per paying member:

	Income earned from dues only, per paying member, per (2) above	14 20	
	Expense, incurred all purposes, per paying member as per (8) above	15 33	
(19)	Excess income earned from dues only over expense incurred all purposes per paying member		1 13

APPENDIX I.

REPORT OF THE PUBLICATION COMMITTEE.

To the Council of the American Society of Mechanical Engineers:

Gentlemen: The Publication Committee would present the following report as work under its direction.

At the close of the fiscal year, September 30, 1902, the Society was under obligation to its printer for work ordered but not completed nor paid for, to an amount of $597.00. This amount has been paid and in addition the expenses for binding and distribution of Volume XXIII for 1901-2 have amounted to $2,757.08.

The net cost of Volume XXIII, was $10,677.54, completed, which makes the cost per copy $4.10. The volume had 878 pages.

The volume of *Transactions* for the current year (Volume XXIV), contains the Proceedings of the New York and Saratoga Meetings. The selection of the month of June for the meeting has made it impossible to bring in as much of the expense of this volume into the current fiscal year as can be done when the meeting falls earlier in the year.

The expense incurred for Volume XXIV to date amounts to $8,656.74 and it is estimated on prices furnished for completed work that a further sum amounting to $6,300 will be needed to complete this volume, making its estimated total cost $14,956.74 as reported in sheet B herewith.

Volume XXIV will contain 1,560 pages, which is about a hundred pages more than the largest previous volume in the Society's history. Its cost has been unusual by reason of the very voluminous contributions to the discussion of the metric system problèm, and the distributions of pamphlet copies to all voters in the membership. The volume will contain, in addition to the papers and discussions, the addresses which were given at the ceremonies connected with the unveiling of the Fulton Memorial in December, 1901. These were omitted from their proper place in the last volume by reason of the necessity imposed for reducing expense at that time. The items which make up the expenditure of the Publication Committee and the totals under each item, are as follows:

Expended for work to date on volume xxiv:

Advance papers	$1,301 07	
Revised papers	1,269 25	
Stenographer's fees	276 75	
Engraving	672 49	
Composition and electrotyping	4,342 48	
Binding extra copies	259 20	
Postage and express	255 50	
Storage	280 00	
Total	$8,656 74—	$8,656 74

Amount brought forward		$6,656 74
Estimated amount required to complete volume xxiv :		
Revised papers, Saratoga	900 00	
Composition and electrotyping	2,400 00	
Binding	2,300 00	
Distribution expenses	700 00	
Total reserved to complete Volume XXIV	$6,300 00—	$6,300 00
		$14,956 74

Respectfully submitted,

PUBLICATION COMMITTEE.

APPENDIX II.

REPORT OF THE LIBRARY AND HOUSE COMMITTEE.

To the Council of the American Society of Mechanical Engineers:

Gentlemen: The Library and House Committee presents the following report of action during the current year. This Committee is intrusted with the functions of control both of the House as the headquarters of the Society and of the Library, which is housed within it.

I. The Library has been open every day between the hours of 10 A.M. and 10 P.M.—excluding Sundays and legal holidays, except during the months of July and August. By reason of sickness in the Library staff the evening openings were suspended during these months. The additions to the Library in the form of exchanges which have been received as the equivalent of the annual volume of the Society's *Transactions* have amounted this year to $533.00.

The Committee has expended for the purchase of books, $130.00, and for binding of periodicals and pamphlets in exchange from other societies the sum of $113.58. There remains a credit to the Society's Library with the house of D. Van Nostrand & Co., for *Transactions* furnished, for which books are to be purchased from that firm, amounting to $328.75. There is a similar credit with Spon & Chamberlain of $16.50. Since the last report a year ago, visitors to the Library have numbered 1,800, averaging six persons a day.

The Library has received from Miss Louisa Lee Schuyler a gift of interesting antiquities from the library of her father, the late George L. Schuyler.

For the conduct of the work in the Library, the Committee has had the services of so much of Mr. Louis A. Gillet's time as could be spared from his duties as Assistant to the Secretary in other lines, and two-thirds of the time, including evenings, of Miss Thornton, as librarian and cataloguer The manuscript of the card catalogue has been supplemented and extended as far as the book titles are concerned, and the Committee hope in the near future to issue a Library catalogue in printed form for distribution.

On account of the expense involved this has not been done up to the present time. The number of volumes in the Library at the date of this report is as follows:

Books	8,500
Pamphlets	3,000

The appraisal reports published on page 19 and 20, of Volume XXIV, made the value of the Library, $10,000; the present book valuation is, $10,979.52. This is based on the additions of new books, and without making allowance for any depreciation.

II. During the fiscal year the Council appropriated for the needs of the house the sum of $5,822.50. The net expense incurred under this appropriation has been as follows:

For operating expenses	$3,347 54
For interest on mortgage	1,402 50
For repairs and additions	644 32
Total	$5,394 36

This is about $200.00 in excess of last year, which is mainly to be attributed to the falling due of long term insurance premiums, and to increased expenditure for necessary furniture. The cost of fuel also this year is in excess of a year ago. The figures in the financial report include credits on House Account, which are not included in the above totals.

The receipts on account of room and hall rentals for the year have aggregated $2,002.25. The total expense of operating the house, exclusive of interest charges on mortgage, repairs and renewals and depreciations has been $3,347.54. Subtracting the receipts makes the net expense of operating the house, $1,345.29, and the total expense, including interest on mortgage, repairs and renewals, and depreciations but excluding an interest on the value of the equity, $5,875.81. The expense incurred in detail for the house has been as follows:

Interest on mortgage	$1,402 50
Gas and electric light	428 40
Fuel	302 50
Janitor's supplies	195 60
Laundry	407 32
Insurance	155 63
Repairs and Renewals, house	298 76
Repairs and Renewals, furniture	345 56
Wages	1,740 00
Incidentals	118 09
Total exclusive of depreciation	$5,394 36
Depreciations	481 45
	$5,875 81

The House Committee employs, for the conduct of the house and library administration, a janitor and his wife (at $60 per month), and has the services for part of his time each day, of a man whose other duties attach to the work of the Secretary's office.

The auditorium has been used during the year for some of the meetings of the Institute of Electrical Engineers, but that organization has outgrown the limited capacity of the hall, and expects to make other arrangements tor the future. The same difficulty has arisen as to the accommodation of the New York Rail-

road Club. The societies meeting regularly in our auditorium now are the Society of Naval Architects and Marine Engineers, the American Society of Heating and Ventilating Engineers, the Society of Municipal Engineers, and a few smaller bodies. A session of the Institute of Mining Engineers was held here in the autumn.

The Committee has considered offers for the House and Lot at No. 12 West Thirty-first Street, ranging between ninety thousand and ninety-five thousand dollars, in view of the inconvenience which would be entailed by present removal from the Society House, and in view of the expected rise in value of the property during the period which must elapse before the Carnegie building is completed, the Committee has reported against the acceptance of these propositions. It has not been thought advisable to raise the appraisal value on the books of the Society.

The Committee believes that, from the location of the house and from the changing character of the street (which is becoming more and more a business centre), the value of its holding will increase as the date of the completion of the Pennsylvania Terminal at Seventh Avenue draws nearer.

Pursuant to the direction of the Council that one per cent. of the total income from dues should be laid aside and reserved for the extension of the Library, the Committee calls attention to the fact reported in the Financial Statement, that the one per cent. for the current year amounts to $319.39. This sum has been deposited in the Institution for the Savings of Merchants' Clerks in New York City, to be drawn upon by the Society, and will be drawing interest until such demand is made.

Respectfully submitted,

LIBRARY AND HOUSE COMMITTEE.

APPENDIX III.

REPORT OF THE EXECUTIVE COMMITTEE.

To the Council of the American Society of Mechanical Engineers:

Gentlemen: The Executive Committee of the Council has special oversight of those expenditures through the Secretary's office, and other channels which do not attach themselves directly to the work of any of the stated committees. The items which fall under the headings of such expenditure for the current year exclusive of salaries, are grouped in the following statement with the amounts attaching to each.

Expense incurred for:

Certificates and introduction cards	$179 11
Badges, distribution and repairs	32 17
Circulars	1,863 02
Meetings	924 92
Catalogues	1,696 86
Office accounts, exclusive of salaries	1,687 00

The account "certificates and introduction cards" covers the expenditure for printing, engrossing and distribution of the diplomas of membership, and the

introduction cards which are given by this Society to members when their initiation fees are paid. The badge account is the expenditure connected with distribution only, since the badge itself is billed to the member at the jewelers' price.

Under the head of circulars the expenditure is grouped into three heads. What are known as "admission circulars" cover application blanks, the confidential inquiries, announcements of election and the like, and have amounted to $580.73. The circulars in connection with meetings are the notices, programs, registers of members in attendance and the like, but does not cover any expense connected with the professional papers. The total this year is $595.23.

Under general circulars are all others which do not fall into either of the other groups. The total this year is $687.06, and is much larger than usual by reason, first of the expenses of printing the draft of the Constitution, By-laws and Rules for the use of the Committee, and for distribution to the members, together with some extra and unusual printing in connection with the expression of opinion which was ordered concerning compulsory legislation on the Metric System.

The Employers' Bulletins issued this year have been four in number, and are included under this heading.

Under the heading of "Meetings" fall the expenses in connection with the two semi-annual meetings, outside of the printing and circulars. Such expenses this year have amounted to $613.20. In addition under the Committee's care have been the monthly reunions of members in New York City, directed by the Council, for which the expenditure has been $311.72. There were four of these meetings held during the months of January, February, March, and April. The topics were:

"The Steam Truck for Heavy Duty," "The Pich Process for Brazing Cast-Iron," "Varnishes," and the "Turbine as a Recorder of the Flow of Streams."

The paper by Mr. Allen on the Turbine has been published in the *Transactions*. The meeting at which the paper on Steam Trucks was presented was the most fully attended.

Under catalogues is included the expenses for composition, press work, paper, and postage, of the two issues of the catalogue. By direction of the Council, these issues were both made this year in the standard size, and the "vest pocket edition" was discontinued. In the July edition a "geographical finding list" was incorporated, and will be continued as a feature of the catalogues as issued in the future. Under the heading of "Office Accounts" are included stationary, postage, telegraph and telephone, office supplies and incidentals. It will be apparent that this group of accounts under the Executive Committee will vary with the size of the Society, and the amounts will increase with the Society's growth.

The Committee would report certain changes under its direction in the matter of salaries paid in the Secretary's office. It has made arrangements whereby the salary paid to the Assistant to the Secretary shall be at the rate of $1,500 for the first year, with an increase of $100 a year to a limit of $2,000. In recognition of the services to the Society rendered by the accountant, and their increasing responsibility as the Society increases, and the amount of income which must pass through his hands, the Committee have recommended that the salary of his position be placed at $2,400 a year.

The great increase in the size of the Society, and the volume of business to be transacted in its offices, has made it necessary to add to the force of stenographers, so that a capable stenographer and typewriter should be available for clerical

work, in addition to the requirements of the correspondence. These changes have been made at different times during the year, so that the total of salaries is different this year from what it will be hereafter when the full yearly rate is to be reported. The expenditures for this year are as follows:

Secretary		$3,600 00
Assistant to Treasurer and accountant		2,300 02
Assistant to Secretary—4¾ months	$950 00	
Assistant to Secretary—7¼ months	928 53—	1,878 53
Stenographer	780 00	
Assistant Stenographer—1 month	30 00—	810 00
Mail clerk		720 00
		$9,308 55

Respectfully submitted,

EXECUTIVE COMMITTEE

SHEET A.

STATEMENT OF CASH ACCOUNT.

FISCAL YEAR 1902–3.

AMERICAN SOCIETY OF MECHANICAL ENGINEERS AND MECHANICAL ENGINEERS' LIBRARY ASSOCIATION.

September 30, 1902, *to October* 1, 1903.

Receipts.

1902. Oct. 1.	To Cash on hand	$83 55
1903. Sept. 30.	" Cash receipts other than Trust Fund Subscriptions during year	46,208 40
"	" Cash receipts, Trust Funds of Mechanical Engineers' Library Association, Fellowship and Sinking Funds of said Association	538 00
		$46,829 95
1903. Oct. 1.	To Cash in East River National Bank	$85 73

Disbursements.

1903. Sept. 30.	By Disbursements for Expenses of Fiscal Year 1901–2		$3,354 08	
"	" Disbursements for Expenses of Fiscal Year 1902–3, all bills received to date paid		32,189 52	
"	" Payment of Note to East River National Bank, N. Y., Liability at end of year 1901–2		8,000 00	
	Total Disbursements for Year			$43,543 60
"	By Money deposited in Savings Banks			
	Trust Fund—M. E. L. A.		$1,481 23	
	Reserve Fund—A. S. M. E.:			
	Library Development	$319 39		
	Life Memberships	400 00		
	Initiation Fees	1,000 00—1,719 39—		3,200 62
				$46,744 22
"	By Cash in East River National Bank			85 73
				$46,829 95

SHEET B.

COMBINED STATEMENT.

AMERICAN SOCIETY OF MECHANICAL ENGINEERS AND MECHANICAL ENGINEERS' LIBRARY ASSOCIATION.

INCOME AND EXPENSE ACCOUNT, *September* 30, 1903. FISCAL YEAR 1902-1903.

Income Earned.

Dues Account—		
Dues collected—Fiscal Year 1902-3....	$31,939 75	
" **Outstanding and considered good, Fiscal Year 1902-3**	1,395 25	
	33,335 00	
Less—1 p. c. carried to Reserve Fund—A.S.M.E.	319 39	
		$33,015 61
Life Memberships	400 00	
Less—100 p. c. carried to Reserve Fund—A.S.M.E.	400 00	
		0 00
Initiation Fees—	5,510 00	
Less—90 p. c. carried to Reserve Fund—A.S.M.E.	4,959 00	
		551 00
Sales Account (charges to sundry purchasers)—		
Publications, except those furnished exchanges for books for Library.... ...	2,327 76	
Publications — Transactions furnished exchanges for books and papers for Library	533 00	
Certificates	50	
		2,861 26
Sales of Electros—Net Gain		4 91
Sales of Second Hand Vols.—Net Gain. ..		115 00
Increased Stock of Transactions on Hand —Inventory, Sept. 30th, 1903	15,607 00	
Less—Inventory, Sept. 30th, 1902	15,495 00	
		112 00
House Account, Rent—		
For Sleeping Rooms	$1,532 25	
" **Hall**	470 00	
		2,002 25

Expenses Incurred.

Transactions, including Distribution—		
Total Disbursements—Fiscal year 1902-3	$11,413 82	
Less—Amount expended to complete Vol. XXIII, the volume for 1901-2, charged against surplus at Sept. 30th, 1902	2,757 08	
Paid on account of Vol. XXIV—Fiscal year 1902-3.	8,656 74	
Estimated amount required to complete Vol. XXIV, for Fiscal Year 1902-3, including cost of Distribution	6,300 00	
		$14,956 74
Office Account—		
Expenses other than Salaries—		
Stationery and Printing	375 97	
Postage, general	639 58	
Telegraph and Telephone	63 73	
Supplies	393 15	
Incidentals	214 57	
	$1.687 00	
Salaries	9.308 55	
		10,995 55
Meetings—		
Annual	$289 97	
Spring	323 23	
Monthly	311 72	
		924 92
Committees		14 67
Catalogues—Composition, Presswork and Paper—January and July, 1903	$1,313 10	
Distribution of both	383 76	
		1,696 86
Circulars, including distribution—		
Admission	$580 73	
Meetings	595 23	
General	687 06	
		1,863 02

	Library—			
	Binding books		$113 58	
	Expense		15 58	
	Salary of Librarian		600 00	
				$729 16
	Certificates and Introduction Cards			179 11
	Distribution and Repairs of Badges—			
	Distribution		$27 17	
	Repairs		5 00	
				32 17
	Interest Paid on Loans			96 19
	Legal Expenses and Expert Fees—			
	Expert Accountant, Auditing Books		$164 37	
	Legal Expenses		54 70	
				219 07
	Uncollectable Accounts—Written off			66 09
	House Account—			
	Lighting	$428 40		
	Fuel	302 50		
	Janitor's Supplies	195 60		
	Laundry	407 32		
	Insurance	155 63		
	Wages	1,740 00		
	Incidentals	118 09		
	Total Cost of Operating House		$3,347 54	
	Interest on Mortgage of $33,000 @ 4½ p.c.		1,402 50	
	Repairs and Renewals—			
	House	$298 76		
	Furniture	345 56		
			644 32	
	Depreciations—			
	House Furniture	$121 45		
	Heating and Vent. App.	360 00		
			481 45	
	Total Expense of House exclusive of Interest on Value of Equity			$5,875 81
				37,649 36
	Balance—Excess Current Income over Expenses incurred for Fiscal Year 1902-3.			1,012 67
Total current income $38,662 03				$38,662 03

SHEET C.

COMBINED BALANCE SHEET.

AMERICAN SOCIETY OF MECHANICAL ENGINEERS AND MECHANICAL ENGINEERS' LIBRARY ASSOCIATION.

September 30, 1903. FISCAL YEAR 1902-3.

Assets.

Property, 12 W. 31st Street, N. Y. Value Appraised	$85,000 00		
Less Mortgage	33,000 00		$52,000 00
Fixtures and Furniture—Heating and Ventilating Apparatus	$3,600 00		
Less 10% for Depreciation for 1902-3	360 00		
Book Value, Sept. 30, 1903		3,240 00	
House Furniture, Book Value, Oct. 1, 1902	$1,214 50		
Less 10% Depreciation for 1902-3	121 45		
	$1,093 05		
Add for New Furniture Purchased 1902-3	112 68		
	$1,205 73		
Deduct for Article Sold	1 47		
Book Value, Sept. 30, 1903		1,204 26	4,444 26
Library, Books, Pamphlets, etc.— Book Value Oct. 1, 1902		$10,315 70	
Additions during the Year 1902-3		663 82	
Book Value, Sept. 30, 1903			10,979 52
Stock of Transactions—Inv. Sept. 30, 1903			15,607 00
Stock of Badges—Inv. Sept. 30, 1903			90 00
Arrears of Dues—Fiscal year 1902-3			1,395 25
Arrears of Trust Funds—Sinking		$10 00	
Fellowship		5 00	15 00
Insurance Premiums Paid in Advance			144 13

Liabilities.

Reserved for Uncompleted Work on Volume XXIV. for Year 1902-3			$6,300 00
Reserved for Interest on Mortgage Accrued			350 62
Advance Payments			197 27
**Library Development Fund*	$319 39		
**Reserve Fund*—Life Memberships	400 00		
† " " Initiation Fees	4,959 00		5,678 39
‡*Trust Fund, M. E. L. A.*			1,496 23
Altoona Mech. Library—Credit Balance with Society			19 25
Surplus, Sept. 30, 1902		$76,423 81	
Deduct Expenditure to complete Vol. XXIII for year 1901-2, now charged against the Surplus at end of that year as a portion of the expense for that year		2,757 08	
		$73,666 73	
Excess of Current Income over Expenses Incurred during Fiscal Year 1902-3 (see Sheet B)		1,012 67	
Surplus, Sept. 30th, 1903			74,679 40

Sundry Debtors—Due the Society :				
For Volumes and Pamphlets		$121 80		
" Room Rent		69 50		
" Hall Rent		35 00		
" Badges		9 00		
" Electros		23 10		
" Gas and Elec. Light—Use of Lantern		2 50		
" Postage and Express		4 50		
			$265 40	
Suspense Account (over-due accounts)—				
For Publications—Volumes and Pamphlets			12 90	
D. Van Nostrand Co.—Exchange Account :				
Balance due Society for Publications			328 75	
Spon and Chamberlain—				
Balance due Society for Publications			16 50	
Engineering Magazine—				
Balance due Society for Publications			1 10	
Stock Second-hand A. S. M. E. Transactions—				
On Hand—Inventory Sept. 30, 1903			135 00	
Union Dime Savings Institution, New York Trust Fund, M. E. L. A.—On deposit		1,481 23		
Institution for Savings of Merchants Clerks, New York—Reserve Fund, A. S. M. E.				
On deposit as follows :				
Life Membership	$400 00			
Library Devl. Fund	319 39			
Initiation Fees	1,000 00	—1,719 39		
Total Funds in Savings Bank			3,200 62	
Cash—East River National Bank, N. Y.			85 73	
			$88,721 16	$88,721 16

* Covered in full by cash in Savings Bank (see Assets).
† Against this there is $1,000.00 on deposit in Savings Bank (see Assets).
‡ Against this there is $1,481.23 on deposit in Savings Bank (see Assets).

Having audited the above Balance Sheet and accompanying Accounts, prepared by Mr. Francis W. Hoadley, with the books and vouchers of the Society, we certify that the same, in our opinion, fully and fairly represent the condition of the Society at September 30, 1903.

NEW YORK, *November* 24, 1903.

SARGENT, PAGE & TAYLOR,
Chartered Accountants.

TABULAR STATEMENT OF CHANGES IN ASSETS AND LIABILITIES,

AMERICAN SOCIETY OF MECHANICAL ENGINEERS AND MECHANICAL ENGINEERS' LIBRARY ASSOCIATION COMBINED.

September 30, 1903, *compared with September* 30, 1902.

Designed and compiled by Francis W. Hoadley, Assistant to Treasurer, and Accountant. Showing whence arise the gains and losses in the financial condition of the Society, as affecting its Assets and Liabilities.

Accounts.	Assets and Liabilities.				Losses and Gains.	
	September 30, 1902.		September 30, 1903.		During Year 1902-3.	
	Assets.	Total.	Assets.	Total.	Losses.	Gains.
Property—12 W. 31st Street, Equity	$52,000 00		$52,000 00			
Fixtures and Furniture—						
Heating and Ventilating Apparatus, Book Value.	3,600 00		3,240 00		*$360 00	
House Furniture, Book Value	1,214 50		1,204 26		†10 24	
Library—Books, Pamphlets, etc., Book Value	10,315 70		10,979 52			$663 82
Stock of *Transactions* on Hand, Inventory Value	15,495 00		15,607 00			112 00
Stock of Badges on Hand, " "	153 00		90 00		63 00	
Arrears of Dues	2,585 34		1,395 25		‡1,190 09	
" " Sinking Fund	50 00		10 00		§40 00	
" " Fellowship Fund	9 00		5 00		§4 00	
Interest on Loans Paid in Advance	96 19		....		96 19	
Insurance Paid in Advance	180 76		144 13		36 63	
Due from Sundry Debtors—Publications	216 35		121 80		94 55	
" " " Hall Rent	25 00		35 00			10 00
" " " Room Rent	21 50		69 50			48 00
" " " Badges	4 50		9 00			4 50
" " " Electros	5 25		23 10			17 85
" " " Miscellaneous	80		7 00			6 20
Suspense Account (Overdue)—Room Rent	120 92		....		120 92	
" " " Publications	75 90		12 90		63 00	
D. Van Nostrand & Co.—Exchange Acct., Balance due Society	287 80		328 75			40 95
Engineering Magazine—Balance due Society	10 50		1 10		9 40	
Spon & Chamberlain " " "	....		16 50			16 50
Stock Second-hand *Transactions* on Hand	....		135 00			135 00
Union Dime Savings Institution, New York, N. Y.—Trust Funds M. E. L. A. on deposit	....		**1,481 23			1,481 23
Institution for Savings of Merchants Clerks, New York, N. Y. Reserve Fund A. S. M. E. on deposit	....		**1,719 39			1,719 39
Cash, East River National Bank, N. Y.	83 55		85 73			2 18
" Fifth Avenue Bank, N. Y.—Trust Fund M. E. L. A.	99 00		....		‖99 00	
Total		$86,650 56		$88,721 16		

	Liabilities.		Liabilities.			
Loans from East River National Bank	$8,000 00		....			8,000 00
Reserved for Interest on Mortgage Accrued	350 62		$350 62			
Advance Payments	177 90		197 27		19 37	
Trust Funds, Mechanical Engineers' Library Association—						
Fellowship	179 00		281 00		102 00	
Sinking	922 23		1,215 23		293 00	
Accounts Payable	597 00		....			597 00
Reserved for Uncompleted Work Volume of *Transactions*	¶2,757 08		6,300 00		3,542 92	
Library Development Fund	....		319 39		319 39	
Reserve Fund A. S. M. E.—Life Membership	....		400 00		400 00	
" " " Initiation Fees	....		4,959 00		4,959 00	
Altoona Mechs. Library—Credit Balance with the Society	....		19 25		19 25	
Total		$12,983 83		$14,041 76		
Surplus, September 30, 1902		$73,666 73				
" " " 1903		74,679 40		$74,679 40		
Total Decrease					$11,841 95	
" Increase					12,854 62	$12,854 62
Net Increase in Assets = Excess of Current Income over Expenses Incurred for Current Year		$1,012 67			$1,012 67	

* 10 per cent. depreciation written off. † 10 per cent. depreciation written off, and amount expended on new furniture added.

‡ Due to collections having been good this year. All of present balance is believed to be collectable.

§ Due to writing off old accounts carried forward from former years but not thought collectable. All the present balance believed to be collectable.

‖ This sum having been transferred to savings bank as a part of the trust fund of the Mechanical Engineers' Library Association now appears in the $1,481.23 to the credit of said fund in savings bank as shown.

¶ Amount paid to complete Vol. XXIII. See Sheet C.

** A total cash asset of $3,200.62 bearing interest in bank has been created this year against the total liability to funds of $7,174.62, *i.e.*,

Liability to Trust Funds Mechanical Engineers' Library Association		$1,496 23	
Cash Asset in Savings Bank			$1,481 23
Liability to Library Development Fund, American Society of Mechanical Engineers		319 39	
Cash Asset in Savings Bank			319 39
Liability to Trust Funds, American Society of Mechanical Engineers—			
From Life Membership Receipts	$ 400 00		
From Initation Fee Receipts	4,959 00		
		5,359 00	
Cash Asset in Savings Bank			1,400 00
Total Liability to Funds		$7,174 62	
Total Asset to Funds (cash in Savings Banks)		3,200 62	$3,200 62
Total Liability to Funds not covered by a Cash Asset in Savings Bank for the reason that the Income of the Year did not allow sufficient Cash available for such purpose		$3,974 00	

ESTIMATED RECEIPTS AND EXPENSES, FISCAL YEAR 1903-4.

October 1, 1903, *to September* 30, 1904.

Estimated Receipts.

Sales Account—Publications		$2,300	
Initiation Fees, 10 p. c. Gross Receipts		300	
Dues, 99 p. c. Gross Receipts		34 000	
House Account—Rent Sleeping Rooms	$1,500		
" " —Rent Hall	300		
		1,800	
Cash on Hand in Savings Bank for Library		319	
Total Estimated Receipts			$38,719
From Reserve Fund *			1,777
			$40,496

Estimated Expenses.

House Account		$3,400	
Interest on Mortgage		1,402	
Repairs and Renewals—House and Furniture		700	
Certificates and Introduction Cards		180	
Transactions—			
Volume 24, cost to complete	$6,300		
" 25, entire cost	11,000		
		17,300	
Circulars		1,700	
Meetings		1,200	
Catalogues		1,700	
Salaries		9,460	
Office Account		1,700	
Distribution of Badges		35	
Library Expense Account		900	
Library Development Account—New Books		319	
Interest on Bank Loans		....	
Committee Work—Carnegie Building		250	
" " —Research		....	
Legal Expenses and Expert Fees—Expert Fees		150	
" " " " " —Legal		100	
			$40,496

* This represents the amount which must be withheld from the annual addition to reserve funds, directed by Council, that one per cent. of the receipts from dues and ten per cent. of the receipts from initiation fees be laid aside: *i.e.*, without such encroachment the reserve is estimated to be $344 + $2,700 = $3,044.

At the conclusion of the presentation of these reports, and some questions and explanations concerning them, the Chair called on Mr. Charles Wallace Hunt, representative of the Society on the Joint Committee intrusted with the consideration of the proposition from Mr. Andrew Carnegie to present a building to the profession of Engineering, for the joint uses of the societies.

The statements made by Mr. Hunt have been embodied in a separate report, which is made an appendix to these minutes, and to which members are referred.

The President then called for the reports of the Tellers, next in order. There were three groups of these reports.

The Tellers of members presented the following report:

REPORT OF TELLERS.

The undersigned were appointed a Committee of the Council to to act as tellers, under Artile 11 of the Rules, to scrutinize and count the ballots cast for and against candidates proposed for membership in their several grades in the American Society of Mechanical Engineers, and seeking election before the XLVIIIth Meeting, New York, 1903.

They have met on the designated day in the office of the Society and have proceeded to the discharge of their duty. They would certify for formal insertion in the records of the Society to the election of the following persons, whose names appear to the appended list in their several grades.

There are 58 members elected, 15 associates, and 46 juniors, making a total increase of 119 names.

There were 567 blue ballots cast of which 12 were thrown out because of informalities. The tellers have considered a ballot as informal which was not endorsed.

C. W. HUNT,
S. S. WEBBER, } *Tellers of Election.*

FOR MEMBERS.

Aiken, Chas. W.	Colwell, J. Van V.	Foucard, M. L.
Albert, Otto	Conrad, E. B.	Gardner, Thos. M.
Albright, H. F.	Cooke, Fred W.	Gilbreth, Frank B.
Allen, Benj. T.	Donnelly, Wm. T.	Goddard, A. L.
Black, Edward S.	Duncan, J. D. E.	Gray, Niel, Jr.
Bloomberg, J. H.	Edgar, Ellis F.	Greene, F. S.
Brown, Hugh T.	Ellicott, Edw. B.	Grossman Albert
Carse, Jno. B.	Fleming, H. S.	Harper, Lewis E.
Child, E. T.	Folger-Osborne, G. F.	Harrington, Jno. L.

Harrington, Norman T.
Hayward, S. F.
Helander, A. H.
Hess, Howard D.
Howe, Albert W.
Hulett, Geo. H.
Ingersoll, Geo. T.
Johnson, Werner
Knox, Luther L.
Lincoln, Robt. B.
Lindstrom, N. O.
MacDonald, D. H.
Merryweather, Geo. E.
Mix, Edgar W.
Moore, A. B.
Moore, Wm. E.
Pattison, F. A.
Peirce, Arthur W. K.
Pritchard, W. S.
Quirk, Wm. M.
Reid, Marcellus
Rickey, Walter J.
Robinson, Frank H.
Shepard, Geo.
Shepard, Louis A.
Sprado, Carl G.
Stebbins, Theo.
Street, Edgar L.
Tandy, Harry
Warg, Robert
Waterman, Charles

FOR PROMOTION TO FULL MEMBERSHIP.

Allan, Percy
Astrom, J. I.
Berg, Hart O.
Blood, Jno. Balch
Bunnell, S. H.
Dollar, Wm. M.
Ducommun, Edward
Ekstrand, Charles
Fernald, Robt. H.
Hoffman, James D.
Jaquays, H. M.
Kellemen, H. F.
Kirk, Robt. H.
Malvern, L. K.
Morrow, Percy C.
Parker, Charles H.
Powell, Emery H.
Rushmore, David B.
Smith, Harry E.
Stanley, A. W.
Stevens, Alfred H.
Trowbridge, Amasa

FOR ASSOCIATES.

Aldcorn, Thomas
Bell, Jno. E.
Caldwell, J. R.
Dornin, Geo. A.
Holz, Fred. H., Jr.
Lauer, C. N.
Loscher, A. P.
Nichols, William W.
Peck, Chas. B.
Pell, David W.
Ransom, Allan
Saldana, E. E.
Saunders, E. W.
Umstead, C. H.
Waddell, Chas. E.

FOR PROMOTION TO ASSOCIATE.

Bateman, Edw. L.
Finley, A. D.
Streeter, L. P.

FOR JUNIORS.

Anderson, H. B.
Atwood, Geo. D.
Bailey, Ervin G.
Barlow, E. S.
Bennett, Geo. G.
Berliner, R. W.
Boughton, J. H.
Brown, E. H.
Case, Albert H.
Chatard, Wm. M.
Colwell, Chas. A.
Coombs, H. A.
Dreyfus, Theo. F.
Ehrmann, Jno. P.
Harvey, Rich. P.
Hawley, Wm. P.
Helmes, M. J.
Henes, L. G.
Howlett, Lewis G.
Jones, Jarrard E.
Jordan, Wm. A.
Jump, E. P.
Kasson, R. S.
Katzenstein, M. L.
Klein, A. W.
Kleinhans, Frank B.
Lowe, Henry L.
Moran, H. P.
Morison, Geo. A.
Morrison, Hunter
Pettit, Frank
Pitkin, Jos. L.
Reis, Leslie R.
Richards, W. A.
Rossberg, Chas. A.
Schuetz, Fredk. F.
Springer, Jno. J.
Staples, H. A.
Stevens, Robt. H.
Stone, Thos. W.
Thatcher, R. P.
Westerfield, Geo. S.
Wettengel, C. Albert
Whittemore, H. L.
Wilson, Henry D.
Woldenberg, I.

The report required no action and was ordered on file.

The tellers to count the letter-ballot concerning the adoption of the proposed Constitution, By-Laws and Rules in the form favorably acted on by the Saratoga Meeting, presented their report as follows:

REPORT OF TELLERS APPOINTED TO COUNT THE BALLOTTS CAST ON THE PROPOSED AMENDMENTS TO THE RULES.

The Committee of Tellers appointed to count the ballots cast by members for and against the adoption of the Constitution, By-Laws and Rules, as reported by the Committee on Revision of the Rules and Methods, begs to submit the following report:

Total ballots cast		448
Ballots thrown out unsigned	4	
" " " signed by rubber stamps	2	
Other informal ballots	0	
Total informal ballots excluded		6
Total ballots counted by tellers		442

Of the regular ballots so counted by the tellers, they would report the following results:

In favor of adoption of said Constitution, By-Laws and Rules	437 ballots
Against the adoption of said Constitution, By-Laws and Rules	5 "
Total	442 "

Our count, therefore, shows that the Constitution, By-Laws and Rules, as submitted at the Saratoga Meeting, 1903, is adopted by the vote of the membership.

Respectfully submitted,

F. M. LAFORGE, } *Tellers of Election.*
C. LOUER, }

On presentation of this report, under the provisions of the new Constitution, the new Constitution, By-Laws and Rules went into effect on the announcement of the president of the formal vote.

The tellers of election of officers presented their report as follows:

REPORT OF TELLERS.

The Committee of Tellers appointed to count the ballots cast by the members for officers of the American Society of Mechanical

Engineers, for the year 1903-1904, begs to submit the following report:

Total ballots cast	702
Ballots thrown out unsigned	23
" " " signed with rubber stamps	0
Other informal ballots (scratched)	21
Total informal ballots	23
Total ballots counted by tellers	702

Of the regular ballots counted by the tellers, they would report the following result:

For President.

Ambrose Swasey, Cleveland	676
Scattering	—

For Vice-Presidents.

D. S. Jacobus, Hoboken, N. J	674
M. L. Holman, St. Louis, Mo	669
Wm. J. Keep, Detroit, Mich	669

For Managers.

George I. Rockwood, Worcester, Mass	672
John W. Lieb, Jr., New York City	673
Asa M. Mattice, Pittsburg, Pa	674
Scattering	1

For Treasurer.

Wm. H. Wiley, New York City	677

Our count shows, therefore, election of Mr. Swasey for President; Messrs. Jacobus, Holman and Keep for Vice-Presidents; Messrs. Rockwood, Lieb, and Mattice for Managers. Mr. Wiley for Treasurer.

Respectfully submitted,

KERN DODGE,
H. M. LANE, } *Tellers of Election.*
S. D. TOMPKINS,

At the conclusion of its reading by the Secretary, the President requested Professor Sweet, appointed a special committee for this purpose, to escort the President-elect to the platform.

Mr. Ambrose Swasey was then greeted by the President and in a few words of recognition and greeting spoke of his pleasure at the honor conferred. He took his seat on the platform at the side of the presiding officer.

The President then called for the regular order of business of the session.

Professor H. W. Spangler presented the report of the committee of which he was chairman, covering the proposed Standard Specifications for Steel Forgings, Steel Castings and Boiler Plate.

This report was discussed by Messrs. Henning, Lanza, Carpenter, Randolph, Kent, Dingee, Flint and Bement.

At the close of the debate on professional questions, Mr. James Christie moved that the Report of the Committee be referred to Committee No. 1 of the American Association for Testing Materials.

This motion being duly seconded was passed. The report and discussion appears as one of the papers of this meeting.

Mr. H. H. Suplee referred to the increasing collection of material of value in the Library of the Society, and that in his investigations as to the very early history of some of the transatlantic societies of engineers it had come to his notice that their early records were very meagre. By the death of those familiar with this early history, these early records had become unattainable, and that while there was a very excellent account of the formation of this Society in the second edition of Volume I of the *Transactions*, there is also a considerable amount of material now extant in the form of recollections of interested members which would in time become unavailable.

Mr. Suplee suggested that the Chair appoint a committee to examine the records of the Society and prepare such formal memorial covering its property and curiosities, as well as recollections of its early history, which should appear either in routine form of the *Transactions*, or separately, as may be thought best, with the particular view of having this material available for the twenty-fifth anniversary of the Society, which would occur within two years.

Mr. Charles Wallace Hunt in seconding this motion spoke of the portrait of John Ericsson, with a very interesting history, and the table of Robert Fulton's, and that in his opinion such a memorial record would be a most interesting and useful record, especially to the younger members of the Society.

The question was put by the President and carried.

The Chair subsequently appointed Messrs. John E. Sweet, Chas. Wallace Hunt and H. H. Suplee.

Professional papers were then taken up for the remainder of the morning:

"Is Anything the Matter with Piece Work"? by Mr. Frank Richards; paper by H. L. Gantt on "Modifying Systems of Management", and by Prof. John E. Sweet on "What are the New Machine Tools to be"? The participants in debate were Messrs. H. L. Gantt, Oberlin Smith, Bates, Fred W. Taylor, Balkwill, DuBrul, Emerson, Parker, Riggs, Fairfield, Henshaw, and Schneble.

The Secretary made some announcements and a recess was taken until the morning of Thursday, December 3rd, at the Steven's Institute of Technology, Hoboken, at 10:30.

Wednesday Afternoon.

For this afternoon the stations of the Interurban Street Railway Company at Ninety-third Street, the Manhattan Elevated Railway Company at Seventy-fourth Street, and the Waterside Station of the New York Edison Company at Thirty-eighth Street, on the east side of the city, were open to members.

The members were assembled at luncheon in the auditorium of the Society House, and parties were made up for the visit to these various points.

No assignment was made for the evening of Wednesday with a view to leaving the visitors free to avail themselves of the opportunities of the city in musical and dramatic lines.

Third Session. Thursday Morning, December 3rd.

By invitation of the President, Trustees and Faculty of the Stevens Institute of Technology, the session this morning was held in the auditorium of the Institute in its main building.

The session was called to order at half-past ten, with President Dodge in the chair. The following professional papers were taken up:

W. B. Gregory, "The Pitot Tube"; F. B. Perry, "Method of Determining Rates and Prices for Electric Power"; D. S. Jacobus, "Tests of a Compound Engine Using Superheated Steam"; E. F. Miller, "The Pressure Temperature Curve of Sulphurous Anhydride (SO_2)"; Benj. T. Allen, "Construction and Efficiency of a Fleming Four-valve Engine"; C. G. Barth, "Slide Rules for the Machine Shop as a Part of the Taylor System of

Management"; and the paper by Mr. F. A. Scheffler on "Suggestions for Shop Construction." The participants in debate on these papers were Messrs. Ennis, Heisler, Suplee, Carpenter, Rockwood, Kent, Kerr, Seymour, Wheeler, Moss, Rice, Child, Goss, Cluett, and R. S. Hale.

At the close of the meeting the members were divided into two groups, one of which took luncheon in the Carnegie Laboratory, and the other was escorted by guides, from the Institute students, through the laboratories and equipment of the Institute.

On completion of their tour, this group was entertained at luncheon, and at half-past two the members re-assembled in the auditorium to listen to a most interesting and striking presentation of the properties of oxides of aluminum in producing intense local heat sufficient to melt masses of considerable size, such as rails, flanges and the like. The most striking example was the melting of a piece of pipe.

Dr. Goldschmidt gave to his lecture the title of Alumino Thermics.

In the evening the usual reception to the visiting members was tendered by the New York members of the Society, at Sherry's, Fourty-fourth Street and Fifth Avenue, New York City.

The reception line included President Dodge and Mrs. Dodge, President-elect Swasey and Mrs. Swasey. Over 600 persons were in attendance and dancing was kept up until a late hour.

Fifth Session. Friday Morning, December 4th.

The closing session of the convention was called to order in the auditorium of the Society, 12 West Thirty-first Street, at ten o'clock. The following professional papers were taken up and discussed: By C. H. Morgan, entitled "A Compact Gas Engine—Beam Type"; "Standard Unit of Refrigeration", by J. C. Bertsch: "A Series Distilling Apparatus of High Efficiency", by Prof. W. F. M. Goss; "Air Motors and Air Hammers, Apparatus and Methods for Testing", by Max H. Wickhorst; "Valve Motion of Duplex Air Compressor", by S. H. Bunnell. The participants in debate on these papers were Messrs. Hobart, Suplee, Jacobus, Voorhees, Bunnell, Shipley, Reeve, Magruder, Kent, Thwait, Morse, Uehling and Jones.

At the close of the professional papers the President announced that new or executive business was in order.

The Secretary by direction of the Council presented the follow-

ing tribute, prepared by Messrs. John E. Sweet, Robert W. Hunt, and John Fritz, a special committee of the Council and which that body had directed should be read in the general meeting.

A TRIBUTE TO DOCTOR THURSTON.

Sudden death has called from us our first President,

DOCTOR ROBERT H. THURSTON,

the one of all best known to us through his work for the Society; the one of all best known in technical education, and the one of all best known to engineers of the world.

His name will last as long as this Society lasts, and we who knew knew him best will hold his name in loving remembrance as long as we live. We mourn him, and to his wife and children, to whom his loss is so much greater, we extend our sincere sympathy.

God has called to Himself one of his noblemen; the peer of his associates and a model for the young men of America.

It is such as he who makes life worth living. Remembering his greatness, let us try to emulate his virtues.

(Signed) JOHN E. SWEET,
ROBERT W. HUNT,
JOHN FRITZ.

The Secretary also reported that during the meeting he had been advised of the death of Sir Frederick Joseph Bramwell, D.C.L., L.LD., F.R.S., Honorary Member of this Society and past President of the Institution of Civil Engineers of Great Britain.

Sir Frederick was apprenticed to a mechanical engineer in 1834, and rose to the position of Chief Draftsman and later to that of Manager. He entered the practice of engineering on his own account in 1853; became an associate of the Institution of Civil Engineers in 1856, a full member in 1862, and President in 1884.

He had also been president of the Institution of Mechanical Engineers in 1874 and 1875, and of the British Association for the Advancement of Science in 1888. He will be remembered by those of the Society who were the guests of the Institution of Civil Engineers in 1899 as one of its most genial and noticeable representatives in the entertainment of the visiting American

engineers. He was as a practitioner especially emphatic as to the dignity and responsibility of the engineer, particularly as owing it to the public that no unworthy schemes reached maturity through lack of exposure of their quality by competent engineers.

The Secretary was on motion directed to transmit a suitable minute and tribute to the Institution of Civil Engineers of Great Britain.

Messrs. Rockwood, Miller, Hunt, Suplee, Goss, Halsey and Engel took part in a somewhat informal discussion as to having the out-of-town membership of the Society represented as bearing a share in the expenses of the annual meeting, recurring each year in the City of New York, and the matter was referred, on motion, to the consideration of the Standing Committee on Meetings created under the new Constitution.

There being no further new or general business to be presented, the President asked Mr. Ambrose Swasey, President-elect, to come to the platform that he might turn over the responsibilities of his office to his successor. Mr. Dodge thanked the Society for the courteous way in which he had been treated during his incumbency, and on motion the meeting adjourned.

On the afternoon of Friday a large number of members accepted an invitation from the De Laval Steam Turbine Company of Trenton, N. J., to be their guests by special train to the works of the Company.

Luncheon was served on the train and the excursion was much enjoyed.

No. 1008.*

CARNEGIE GIFT TO ENGINEERING.

SECOND CIRCULAR.

At the annual meeting of the Society held in New York City, December 1st—4th, a report was called for from the representative of the Society on the Joint Committee intrusted with the details of Mr. Andrew Carnegie's proposed gift of a building for the needs of the engineering societies.

Mr. Charles Wallace Hunt, as such representative, presented a verbal report, from which the following information is derived.

It will be recalled that in advance of the Saratoga Meeting an announcement was made to the Society, which will be found at page 870, Volume XXIV, of the *Transactions*. It mentioned the purpose of Mr. Carnegie's donation of one million dollars, which should bring the libraries, assembly-halls, offices and meeting rooms of the societies of engineers into one great building, which, while ample in size for their individual needs, should arrange for their convenience as respects business and professional uses. It was proposed, in addition, that the building should give adequate accommodation for such other technical, scientific and engineering bodies as might require the use of a properly equipped auditorium.

During the intervals between that report and the present meeting, the Joint Committee of fifteen has been in conference on the details referred to it, and has appointed an Executive Committee of five to facilitate the work of the general committee.

A sub-committee of the Executive Committee has been engaged in the preparation of the material which might be necessary to submit the needs of these societies to a competition of architects, as to the structural details of the building. A Committee on Organization has been appointed, consisting of Messrs. A. R. Ledoux, Chas. Wallace Hunt and S. S. Wheeler, and their

---

* Appendix to Proceedings.

recommendations to the Joint Committee have been adopted and the Organization Committee instructed to take the necessary preliminary steps to secure the legislative action for which their report called.

The full report of the Committee on Organization is as follows:

The undersigned were appointed a special Committee at a meeting of the Executive Committee of the Union Engineers' Building Association, on July 9, 1903, and were instructed 'to propose a plan of organization for the bodies exclusive of the Engineers' Club, which are to participate in the gift.' In transmitting instructions to this Committee, the Chairman of the Executive Committee summarized our duties, as he understood them, as follows:

1. To suggest an organization to hold title to the land and building.
2. A method of superintending and administering the building after completion.
3. Provision for granting the use of parts of the building to other organizations whose objects and work may be of suitable character and which may contribute to the maintenance of the building.
4. Provision for contingencies, such as the withdrawal of one of the societies, or the failure on the part of any one properly to carry out its obligations.

The Committee was authorized to consult legal counsel. It has held several meetings and has informally consulted members of the American Society of Civil Engineers, and others interested.

After having informally and tentatively approved suggestions formulated by one of its members, Mr. Hunt, the same were submitted to the law firm of Butler, Notman, Joline & Mynderse, from whom a written opinion was obtained.

Having considered this legal advice and taken note of all suggestions received, the Committee unanimously advise as follows:

1. The total amount offered by Mr. Carnegie shall be administered as two gifts; one to the Engineering Societies and the other to the Engineers' Club, each to be held and administered independent of the other. The allocation of the fund to be made at once, but the buildings to be designed and erected as one operation; thereafter the respective titles and administrations to be entirely independent.
2. The property represented by land, buildings and equipment of the engineering societies, shall be held and administered by an

executive corporate body, preferably under a special charter, to be obtained from the State of New York, each of the constituent societies being entitled to name from its membership three persons to act as incorporators and thereafter as directors.

3. Each society annually to elect or appoint, as their By-Laws may prescribe, one of their voting members to serve on the Board of Directors of the Executive Corporation for a term of three years; a vacancy in said Board to be filled by an appointment made by the society, the retirement of whose representative causes the vacancy.

4. The land and property being held for the societies by an executive corporation, the said corporation may, to pay for the land acquired, issue certificates of indebtedness or bonds bearing interest at four per cent., and redeemable on six months' notice, the buildings being a gift from Mr. Carnegie.

5. Each of the constituent societies may purchase and hold an equal amount in value of the said bonds or certificates, but the Board of Directors of the executive corporation may authorize any of the constituent societies to hold an additional amount; that is in excess of its portion, but such excess shall be subject to recall at its par value at any time that the directors of the Executive Corporation may so order, to the end that each society shall have an equal interest in the property of the corporation if it so desires.

The certificates held by each society shall be inalienable unless they are offered to the Executive Corporation at their par value, and such tender shall not be accepted by the Board of Directors within one year thereafter.

6. The property of the Executive Corporation shall be used perpetually as a meeting-place and headquarters for the constituent societies, and for such other scientific associations as may be temporarily admitted by the consent of the Board of Directors of the Executive Corporation. Such associations may pay a pro rata share in the expenses of the headquarters, but no profit shall be made from such use.

7. Each of the participating societies shall be entitled to rooms and space in the property adequate to its need, paying its share of the running expenses in accordance with the amount of space occupied; said space to be assigned and a proper assessment therefor determined by the Board of Directors of the Executive Corporation.

8. The excess of receipts over expenditures, if any, shall be used for reducing the subsequent contribution of the several societies for maintaining the building, and for the advancing of engineering arts and science, by and through the participating constituent associations. No dividends shall be declared or profits divided, but a reasonable repair and rebuilding fund may be established.

9. If the income of the Executive Corporation shall be less than

the expenditure, the deficiency shall be made good by an assessment on each of the constituent societies, so allocated as to be in proportion to the number of voting members of each society.

An excess of receipts over expenditures may be allocated to the societies in like manner to reduce their annual assessment.

10. Should any of the constituent societies fail or refuse to appoint directors, the remaining members of the Board of the Executive Corporation shall administer the property with all the force and effect as though the Board contained its full quota of members.

11. Finally, your committee, in offering the above suggestions, has had in mind the setting aside of the money used for a building for the Engineers' Club, so that on the completion of the said building, the relations of the Club and of the engineering societies will terminate. Thenceforward, the constituent societies are to carry through the Executive Corporation the administration of the building and its accessories, leaving the scientific, professional, intellectual and financial activity in each organization entirely independent of the others, and free to develop to any extent and along any line that may be determined each for itself.

The details of the superintendence and administration of the buildings can best be considered after the organization of the proposed Executive Corporation through the procuring of a special charter.

(Signed) ALBERT R. LEDOUX,
CHAS. WALLACE HUNT,
SCHUYLER S. WHEELER, } *Committee on Organization.*

It may be mentioned that when the sub-committee on building plans took up the consideration of the necessary areas it developed at once that the cost of the building which would be required for the accommodation of the Engineering societies, on 125 feet front on Thirty-ninth Street, and the Engineers' Club on a fifty foot front on Fortieth Street, would exceed the preliminary or tentative figure mentioned by Mr. Carnegie in his official letter of gift.

When this conclusion was reached, a special committee was appointed to visit Mr. Carnegie and talk the matter over with him. He made it evident that it was his desire and intention to erect a building which should be a monument to the Engineering profession, and that it was his purpose to make this building adequate and the question of cost was in his mind secondary to the attaining of the object dear to his heart, which was the bringing to-

gether of all classes of engineers into a building in which they could co-operate for the common purpose of the profession.

It may be desirable to repeat some of the statements of the previous announcement, that the location of the proposed building, as given in the sketch plan herewith, is one of the most desirable ones in the city of New York.

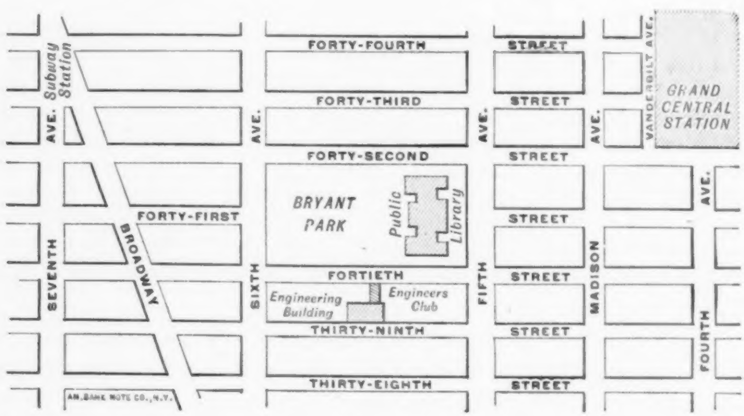


FIG. 1.

The New York Public Library building occupies the entire eastern frontage of the block between 40th and 42nd Streets on Fifth Avenue, while to the westward is the open square known as Byrant Park.

It will be conveniently accessible from the terminals of the New York, New Haven and Hartford Railroad, and the New York Central and Hudson River Railroad at 42nd Street, at Park and Madison Avenues, and from the Pennsylvania terminals coming in from the West, at 33rd Street and Seventh Avenue. The Elevated railway at Sixth Avenue has a station at 42nd Street, and the Third Avenue Elevated, 42nd Street and Park Avenue. The junction of the Rapid Transit underground lines at 43rd Street and Broadway will make this station an important express point for trains both from down town and from the North.

Convenient surface lines, both up and down town, intersect at 42nd Street. The hotel and theatre district of the city are centering more and more around this particular region.

The Committee will be very glad to receive from members of

the Society suggestions which may be helpful in the matter of interior arrangement or detailed requirement of the proposed building, to the end that it may be in every respect thoroughly adapted for the needs of the societies which are to occupy it.

The committee would report that for the three societies which have so far signified their purpose to make use of the proposed building, the requirement of floor space in square feet, outside of the general rooms for auditoriums and library needs, is as follows:

Purpose.	Electrical.	Mechanical.	Mining.
Reception room	600	800	620
Editorial room	400	300	360
Secretary or Assistant	400	300	360
Counting room	600	600	320
Accountant—Stenographers	400	900	700
Board and Committee	1,000	600	760
Stationery and Transactions	1,200	1,200	1,000
Closets	400	400	400
Sundries	200	400	600
	5,200	5,500	5,120

The Society of Civil Engineers has issued a circular to its members reporting the condition of affairs at the present time and their probable need for 9,000 square feet of floor space, but definite action cannot be taken by that society until a letter-ballot is had and until after the annual meeting of the society in January.

In the letter-ballot taken on the question of the necessary changes in their By-Laws by the American Institute of Mining Engineers, the vote showed over 1,600 in favor of the change, and only eleven in opposition to it, which is regarded as showing an overwhelming sentiment in favor of the Engineering Building.

It may be further of advantage to report that the action in the Joint Committee on all questions which have been submitted to it has been unanimous, and that throughout, in all decisions, the Joint Committee has had the benefit of the counsel, advice and co-operation of the representatives appointed by the American Society of Civil Engineers, who have sat with the Committee in all its deliberations.

Respectfully submitted,

JAMES M. DODGE,
C. W. HUNT,
F. R. HUTTON,
Committee.

No. 1009.*

THE MONEY VALUE OF TECHNICAL TRAINING.

BY JAMES M. DODGE, PHILADELPHIA, PA.

PRESIDENT'S ADDRESS, 1903.

1. Technical Training may be self-acquired or obtained through instruction. The ability to drive a nail properly, or to design and construct the most complex and wonderful of structures or devices, is the result of Technical Training in but different degree. Up to a very recent date, and within the memory of most of us, the Apprentice System and that of Independent Delving represented the sole methods of acquiring training. Research and investigation carried on in individual lines, with varying degrees of success, dependent upon the mental makeup of the individual, were the means of attaining theoretical technical knowledge. The blending of these two methods developed the earlier Mechanical Engineers and will, even in the future, enable those sufficiently gifted by nature and habit to attain eminence. The progress of the world, however, calls for a better and more speedy means of producing trained men than could ever be developed by the methods of self-instruction. The individual, striving for manual skill, attains his desire under the old apprentice system. Individuals sufficiently gifted arise above their fellows, and become the leaders in their calling. The gratification of a mechanical appetite and the desire to earn more money than his fellows are two moving causes which impel a man towards technical education. A generation or so ago, the universal belief was that the sooner a young man entered upon his apprenticeship, or began practical manual work, the better and more rapid would be his progress in the Mechanic Arts, and Book Learning was derided as being purely theoretical, and of little practical value.

* Presented at the New York meeting (December, 1903) of the American Society of Mechanical Engineers, and forming part of Volume XXV. of the *Transactions*.

This belief is, even at this date, all too prevalent, largely due to inherited error, and to lack of knowledge and reliable data.

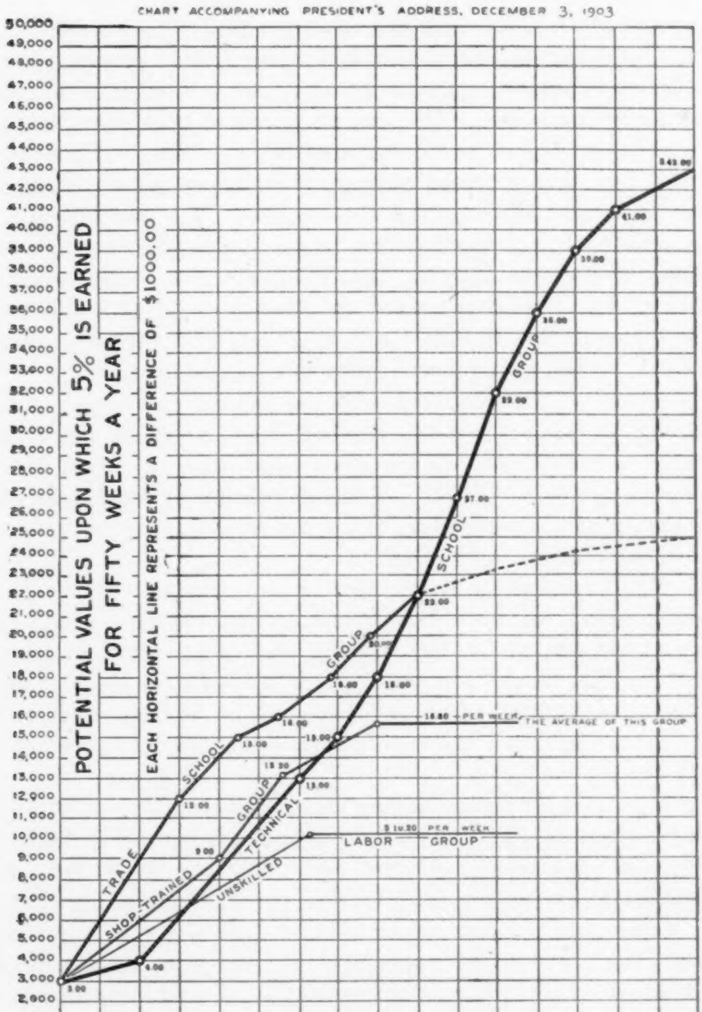
2. Obtaining data from which incontrovertible conclusions can be drawn is now comparatively easy, but a few years ago was practically impossible. We are all prone to take extreme cases of success or failure as the basis of our opinions, and lose sight of the fact that it is the average man whose career shows the true force and direction of the current. For convenience of comparison, I will outline the actual progress made by four groups of men working in the Mechanic Arts,—the unskilled labor group, the shop-trained or apprentice group, the trade school group, the technical school group, and give the results attained. Each group I will refer to as an individual:

3. The first is the Laborer, with but primitive and rudimentary training, working under the immediate and constant supervision of a boss, and earning, as the line on the chart indicates, $10.20 per week at the age of 22, his line remaining horizontal through the period of his usefulness. Data are lacking as to his progress before he reaches the age of 22.

4. The second is the Apprentice or Representative of the Shop-trained group, of good health and habits, entering a machine shop at the age of 16, and earning an average wage of $3 per week for fifty weeks per year. This is about the number actually worked, making $150, or 5 per cent. on $3,000, which is his Potential or Invested Value, upon which he draws his interest on pay days.

5. On the chart accompanying this paper (Fig. 2) you will find, ruled horizontally, lines representing amounts increasing from the lower line upward by $1,000 each; starting at $1,000, and terminating at the top at $50,000, these representing Potential Values, upon which 5 per cent. is earned for fifty weeks a year. The vertical lines each represent one year in time, beginning at the lower left hand corner at 16, and progressing in regular order until, at the lower right hand corner, we have 32, representing in all a lapse of 16 years.

6. To illustrate the progress of the four groups graphically, we indicate on the line representing 16 years of age, and opposite the figure $3,000, the young man just entering his apprenticeship. We will consider him typical of the Shop-trained Group. Following the line to the right we see his average progress in earning capacity through the ensuing years, noting that at the



THE MONEY VALUE OF TECHNICAL TRAINING

Fig. 2.

age of 20 he is earning \$9 per week, which is 5 per cent. on \$9,000, he having increased his Potential or Invested Value in four years by \$6,000.

7. We now note that his accumulated experience enables him to make more rapid progress for the next year and a half, and from the age of 20 to 21½ years we find that his pay has increased to \$13.20 per week, and his Potential Value to \$13,200. He is now approaching his goal, and his line of progress does not continue at the same angle that it followed for the past few years, but deflects toward the horizontal; and at the age of 24 we find him earning \$15.80 per week, and his Potential Value \$15,800. In other words, in eight years he has increased his Potential Value \$12,800. Observation shows that 5 per cent. of the apprentices acquiring the machinist trade rise above the line made by our average man; 35 per cent. follow the line closely, and that during the period of training 20 per cent. leave of their own accord, and as near as can be ascertained, go to other shops and continue in the line originally selected; 40 per cent., however, are found unworthy or incompetent, and are dismissed, probably never rising to the \$15.80 line.

8. Apprenticeship of to-day in many establishments does not make the man, broadly speaking, a mechanic—in a majority of cases he is a specialist or tool hand, and not comparable with the old mechanic, who was a worker in metals, had some practical knowledge of steam and prime movers, could chip, file, work on lathe, planer, drill press, or as an assembler, and was competent to meet the varied and unusual conditions found in general construction and repair work.

9. The third group of young men are those fortunate enough to have had the opportunity of entering a trade school, which they do at 16 years of age, devoting the next three years of their lives, or until they are 19 years of age, acquiring a trade under competent instruction, and at the same time adding to their store of rudimentary theoretical education. At the age of 19 a Trades' School man enters the machine shop and can command \$12 per week, equal to the apprentice at 21 years of age, and very quickly makes his employment profitable to his employer. The three years in school have increased his Potential Value from \$3,000 to \$12,000, a gain of \$9,000. Thus he has caught up with the apprentice entering the shop at 16, and who has been working for five years. Progress of the Trades' School group now follows

a line which diverges from that of the regular apprentice, and by the time $15.80 is earned by the regular apprentice, the Trades' School graduate is earning $20, with a Potential Value of $20,000, or $4,200 greater than that of the Shop-Trained man. The Trades' School line continues at substantially the same angle up to an earning capacity of $22 per week, and a Potential Value of $22,000. Data are lacking as to the further progress, but the presumption is that this line will bear off more toward the horizontal, eventually paralleling the line of the Shop-trained man, but much higher on the chart.

10. The fourth group we will represent again by a boy of 16 studying at school until his 18th year, and preparing himself for admission to one of our higher Institutions of Technical learning, such as the Stevens Institute, the Massachusetts Institute of Technology, Columbia, Cornell and the like, where, after a four years' course, or at the age of 22, he is ready to begin practical work. The statistics upon which this chart is based show the average starting wage at $13 per week, or the same amount earned by the regular apprentice at the age of 21½, and by the Trades' School graduate at the age of 19½. In other words, apparently a graduate of our technical schools has lost by his six years of preparatory study, having been beaten by the regular apprentice by six months and by the Trades' School graduate by 2½ years. From this time, however, there develops a most interesting and instructive line of progress. The regular apprentice, who is earning $13.50 a week at the time the technical graduate is earning $13, is overtaken in six months, and we find both earning $14 per week, and the technical graduate reaches the $15.80 line nearly one year before the regular apprentice. In other words, while it has taken the regular apprentice from his 21st to his 24th year, or three years, to increase his wages from $11.50 to $15.80 a week, the technical graduate has done the same in fifteen months.

11. Progress now continues on substantially the same line, and we find the technical graduate earning $22 per week, and crossing the line of the Trades' School group in three years' time, a worthy tribute to the higher education and attainment.

12. The line of the technical graduate now continues divergent from that of the trades' school graduate, with earning capacity regularly increasing, and a corresponding augmentation of Potential or Invested Value until, at the age of 32, or ten years

after entering upon the practical work, we find our technical graduate earning $43 per week, and his Potential Value at $43,000. In other words, six years of preparation have enabled him to distance the Shop-trained man and the Trades' School graduate overwhelmingly. Bearing in mind that this is an average line, it is of interest to say that most technical graduates with a better record than the one in the chart have devoted even more time to their preparation, either by study or by shop work, after graduation. Those, on the other hand, who have not come up to this average line represent, in the main, men more or less incapable of original work. The reason that higher education, other things being equal, carries with it the ability to earn high wages is that consciously or unconsciously, these men are directing and making it possible for large numbers of Laborers, Shop-trained men and Trades' School graduates to perform useful work. A draftsman at his board may never realize that as a result of his drawing a hundred men or more may be given employment. His design calling for structural steel, for instance, could not be built were it not for the labor of many men employed making and rolling the steel before it reaches the shop. Then come the shop men, who cut, punch and shear, and then the erectors, who assemble the structure in accordance with the original plan. For this ability and knowledge our technical man is paid.

13. It is quite obvious that all workers in the Mechanic Arts cannot be technical graduates. Some must, through natural limitations, or lack of opportunity, follow the apprentice line, and others the Trade School.

14. It is from graduates of the latter that leading shop men and foremen are largely selected. These two classes, supplemented by the technical graduate, constitute the vast army of workers in the Mechanic Arts.

15. Thus we see clearly that preparation pays, and that it pays in dollars and cents, and that even a long term of years spent in proper study and technical training is a good investment from every point of view.

16. Of course, apprentices have made and will make, in rare instances, a better showing than the average technical man of the chart, and many of our greatest men have, by sheer force of character, excellence of brain fibre, persistence and self-education, risen to preëminent positions, independent of all regular

systems. To the end of time great examples of this kind will be found. Among those whose names readily come to mind are the elder Krupp, Joseph Whitworth, George M. Pullman, Andrew Carnegie, John Fritz, Prof. John E. Sweet, Edwin Reynolds, George H. Babcock and Coleman Sellers.

17. The same is true of the Trades' School graduate, but as said before, we are dealing with the average of each class, taken from actual statistics, with an earnest desire to ascertain the facts, and without any preconceived notion of the outcome.

18. It may be stated as a truism that every man pays for the amount or percentage of bossing he requires, and conversely, every man's wages increase in proportion to his ability to act as the boss or foreman of himself and others. The lower the wage rate the greater the amount of watching and directing constantly required. The slaves of ancient Egypt received no wages, but were treated as horses are to-day. They were fed and sheltered according to the ideas of their owners. No slave worked voluntarily, and the foreman's or leader's excellence was gauged entirely by his physical strength and efficiency as a driver. This was certainly the zero of labor conditions.

19. The highest wages are paid to the man through whose ability the largest number of other men may be most profitably employed. He does his work with his brain. Thus, on the one hand, we see manual labor receiving no wages, and on the other mental labor reaping the highest reward. Between these two extremes is found every condition of human life.

20. A practical man performs his work within the radius of his arm, a technical man within the radius of his brain. This fact is, even to-day, realized by the few, but it is gratifying to know that the number is increasing.

21. The technical training of an individual makes him valuable just in proportion as his ability is manifested by good judgment and perception. Trained common sense receives the highest compensation and reaps the greatest reward.

22. Mental ability to receive ideas and impart them properly and wisely, rearranged and grouped, is typical of the most brilliant mentality; a dull intellect may be compared to blotting paper, fit only to absorb and inter a heterogeneous mass of impressions.

23. The most interesting of all graphical charts would be that properly exploiting the value of technical training to manufacturing plants and enterprises. To illustrate this more clearly,

we may fairly assume that the apprentice of our chart corresponds to the old-fashioned primitive shop, having practically no overhead expense, the proprietor carrying the business "in his hat," priding himself on his non-receptive sturdiness, contempt for improvements and personal attention to all details. For his costs he adds together the value of raw materials and labor, and then adds a few dollars for profit. The line of this establishment would parallel the $15.80 line of our Shop-trained group.

24. The Trades' School line on the chart truthfully represents establishments in which some attention has been paid to the improvement of system, with an increased so-called non-productive force, operating possibly in some particulars with brilliancy, but with defective features in others; acknowledging the value of improvement if internally originated; moderately but unconsciously absorbent of ideas from without, but tenacious of dogma and lacking departmental symmetry. Growth, increased earnings and relative immunity from disastrous failure result.

25. The technical graduate line of our chart represents the manufacturing establishment technically trained and "abreast of the times" in all particulars, and I predict a time not very far distant when it will be almost universally recognized that establishments should be trained as well as individuals, and that the marvellous development in scientific shop practice and management will do for the manufacturer fully as much as technical training is doing for the individual.

26. A change of mental attitude towards the subject of advanced Shop Practice and Management is noticeable to a marked degree. Within a very few years, indifference and antagonism have changed to a growing interest and appreciation.

27. The greatest musical composition contains no new notes; each note of the scale can be sounded on a penny whistle. Our greatest composers have only arranged the notes in harmonious sequence. The artists that can render their music truly, well deserve unstinted praise, even though they lay no claim to the composition of the masterpiece. Truly a listener at the grand opera could say, "There is nothing novel in this; I have heard every one of these notes before. I have even made similar sounds myself, and the result was far from satisfactory." So with Shop Management: it must be as fundamentally harmonious as a musical composition, and need not of necessity embody within it any one element of extreme originality. Of it the individual

may truly say, "Nothing novel has been presented; I tried this feature or that feature with no beneficial result," but if he can play the music of the Art of Management thoroughly well he need not grieve because he is not its composer.

28. Henry R. Towne, F. A. Halsey, H. L. Gantt and Charles Day have all ably contributed through our Proceedings to the literature of this most important subject. Fred. W. Taylor, in his paper of the current year, while claiming no originality of detail, has presented to the world the most complete and thoroughly scientific system of shop management ever promulgated. As an investigator and student he is sowing seeds in the field of the Mechanics Arts which will bear a bounteous harvest.

29. It may be truly said that this Society, and others allied in promoting the Mechanic Arts, complete the system of technical training by going beyond the province of the technical schools, their students being the men who constitute the management of our manufacturing enterprises. It should be gratifying to all of us that the pioneer literature of advanced shop management and practice for this post-graduate course of technical training was presented to the world by the American Society of Mechanical Engineers.

No. 1010.*

SLIDE RULES FOR THE MACHINE SHOP AS A PART OF THE TAYLOR SYSTEM OF MANAGEMENT

BY CARL G. BARTH, SWARTHMORE, PA.

(Member of the Society.)

1. In his paper on "Shop Management," read at the Saratoga meeting of the Society in June last, Mr. Fred W. Taylor referred to certain slide rules that had been invented and developed under his supervision and general guidance, by means of which it becomes a comparatively simple matter to determine that feed and speed at which a lathe or kindred machine tool must be run in order to do a certain piece of work in a minimum of time.

2. These slide rules were also mentioned by Mr. H. L. Gantt in his paper "A Bonus System of Rewarding Labor" (New York Meeting, December, 1901), as being at that time in successful use in the large machine shop of the Bethlehem Steel Company, and reproductions of a number of instruction cards were therein presented, the dictated feeds and speeds of which had been determined by means of these slide rules.

3. Mr. Taylor early set about making experiments with a view to obtaining information in regard to resistances in cutting steel with edged tools, and also the relations that exist between the depth of cut and feed taken to the cutting speed and time that a tool will endure; and he advanced far enough along these lines in his early position as engineer for the Midvale Steel Company to make systematic and successful use of the information obtained; but as this, of course, was confined to tempered carbon tools only, it was not applicable to the modern high-speed steel, so that the invention and introduction of this steel called for new experiments to be made.

4. These were first undertaken under Mr. Taylor's directions at

* Presented at the New York meeting (December, 1903) of the American Society of Mechanical Engineers, and forming part of Volume XXV. of the *Transactions.*

Bethlehem, so far as the cutting of steel alone was concerned; and later on at the works of William Sellers & Co., Inc., of Philadelphia, at which place the writer spent fifteen months in going over these experiments again, on both steel and cast iron, and with tools of a variety of shapes and sizes, and for which nearly 25 tons of material were required.

5. However, it is not the writer's intention at this time, to give an account of these experiments, or of the results obtained and conclusions drawn from them, but merely to give some idea of the slide rules on which these have been incorporated, and by means of which a most complex mathematical problem may be solved in less than a minute.

6. He will also confine his attention to the most generally interesting of these slide rules; that is, the slide rules for lathes, and he will take for an example an old style belt-driven lathe, with cone pulley and back gearing.

7. Considering the number of variables that enter into the problem of determining the most economical way in which to remove a required amount of stock from a piece of lathe work, they may be enumerated as follows:

I. The size and shape of the tools to be used.
II. The use or not of a cooling agent on the tool.
III. The number of tools to be used at the same time.
IV. The length of time the tools are required to stand up to the work (LIFE OF TOOL).
V. The hardness of the material to be turned (CLASS NUMBER).
VI. The diameter of this material or work.
VII. The depth of the cut to be taken.
VIII. The feed to be used.
IX. The cutting speed.
X. The cutting pressure on the tool.
XI. The speed combination to be used to give at the same time the proper cutting speed and the pressure required to take the cut.
XII. The stiffness of the work.

8. All of these variables, except the last one, are incorporated in the slide rule, which, when the work is stiff enough to permit of any cut being taken that is within both the pulling power of the lathe and strength of the tool, may be manipulated by a person who has not the slightest practical judgment to bear on the matter;

but which as yet, whenever the work is not stiff enough to permit of this, does require to be handled by a person of a good deal of practical experience and judgment.

9. However, we expect some day to accumulate enough data in regard to the relations between the stiffness of the work and the cuts and speeds that will not produce detrimental chatter, to do without personal judgment in this matter also, and we will at present take no notice of the twelfth one of the above variables but confine ourselves to a consideration of the first eleven only.

10. Of these eleven, all except the third and tenth enter into relations with each other that depend only on the cutting properties of the tools, while all except the second, fourth and ninth also enter into another set of relations that depends on the pulling power of the lathe, and the problem primarily solved by the slide rule is the determination of that speed-combination which will at the same time most nearly utilize all the pulling power of the lathe on the one hand, and the full cutting efficiency of the tools used on the other hand, when in any particular case under consideration values have been assigned to all the other nine variables.

11. If our lathe were capable of making any number of revolutions per minute between certain limits, and the possible torque corresponding to this number of revolutions could be algebraically expressed in terms of such revolutions, then the problem might possibly be reduced to a solution, by ordinary algebraic methods, of two simultaneous equations containing two unknown quantities; but as yet no such driving mechanism has been invented, or is ever likely to be invented, so that, while the problem is always essentially the solution of two simultaneous equations, or sets of relations between a number of variables, its solution becomes necessarily a tentative one; or, in other words, one of trial and error, and involving an endless amount of labor, if attempted by ordinary mathematical methods; while it is a perfectly direct and remarkably simple one when performed on the slide rule.

12. The slide rule method of solution may, however, also be employed for the solution of numerous similar problems that are capable of a direct and perfect algebraic solution; and it will, in fact, be best first to exhibit the same in connection with the simplest imaginable problem of this kind.

13. In the first place, the solution of two simultaneous equations may be graphically effected by representing each of them by a curve whose coördinates represent possible values of the two

unknown quantities or variables, for then the coördinates of the point of intersection of these curves will represent values of the unknown quantities that satisfy both equations at the same time.

14. *Example* 1. Thus, if we have $y + x = 12$ and $y - x = 3$, these equations are respectively represented by the two straight lines AB and CD in Fig. 3; and as these intersect at a point (1)

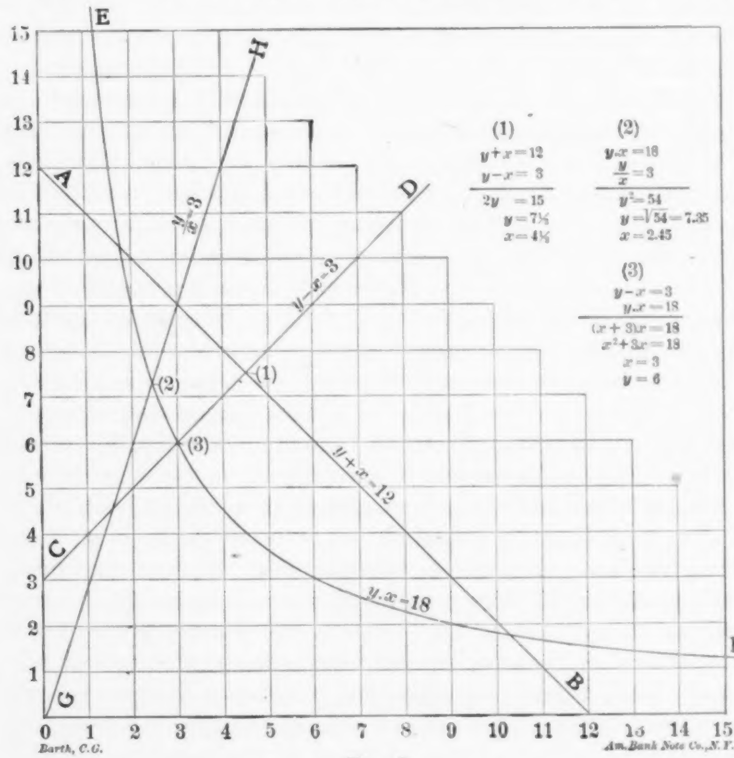


FIG. 3.

whose coördinates are $x = 4\frac{1}{2}$ and $y = 7\frac{1}{2}$, these values will satisfy both equations at the same time.

15. *Example* 2. Suppose again that we have $x.y = 18$ and $\frac{y}{x} = 3$, and these equations are respectively represented by the equilateral hyperbola EF and the straight line GH; and the coördinates to the point of intersection of these (2) being respectively $x = 2.45$ and $y = 7.35$, these values will satisfy both equations at the same time.

16. *Example* 3. Similarly, if we have $y - x = 3$ and $y.x = 18$, these equations are respectively represented by the straight lines *CD* and the equilateral hyperbola *EF;* and the coördinates to the point of intersection of these (3) being $x = 3$ and $y = 6$, these values will satisfy both equations at the same time.

17. The slide rule method of effecting these solutions—to the consideration of which we will now pass—will readily be seen to be very similar in its essential nature to this graphical method, though quite different in form.

18. In Fig. 4 is shown a slide rule by means of which may be solved any problem within the range of the rule of the general form: "*The sum and difference of two numbers being given, what are the numbers?*"

19. The rule is set for the solution of the case in which the sum of the numbers is 12 and their difference 3, so that we may write

$$y + x = 12 \text{ and } y - x = 3,$$

which are the same as the equations in Ex. 1 above.

20. In the rule, the upper fixed scale represents possible values of the sum of the two numbers to be found, for which the example under consideration gives $y + x = 12$, opposite which number is therefore placed the arrow on the upper slide.

21. The scale on this slide represents possible values of the lesser of the two numbers (designated by x) and the double scale on the middle fixed portion of the rule represents possible values of the greater of the two numbers (designated by y); and these various scales are so laid out relatively to each other, and to the arrow referred to, that any two coincident numbers on these latter scales have for their sum the number to which this arrow is set; in this case accordingly 12.

22. The bottom fixed scale on the rule represents possible values of the difference of the two numbers, in this case 3, opposite which number is therefore placed the arrow on the bottom slide of the rule, the scale on which also represents possible values of the lesser of the two numbers, x; and the double fixed scale in the middle of the rule representing, as already pointed out, possible values of y, the whole is so laid out that any two coincident numbers on these latter scales have for their difference the number to which this arrow is set; in this case accordingly 3.

23. Fixing now our attention on any number on the double y scale in the middle of the rule, we first note the values coincident

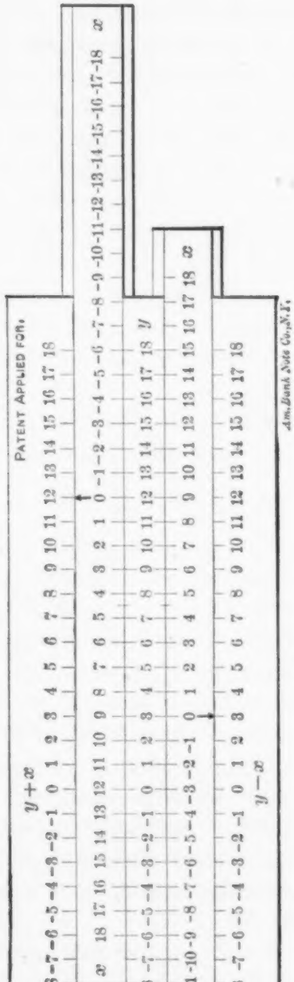
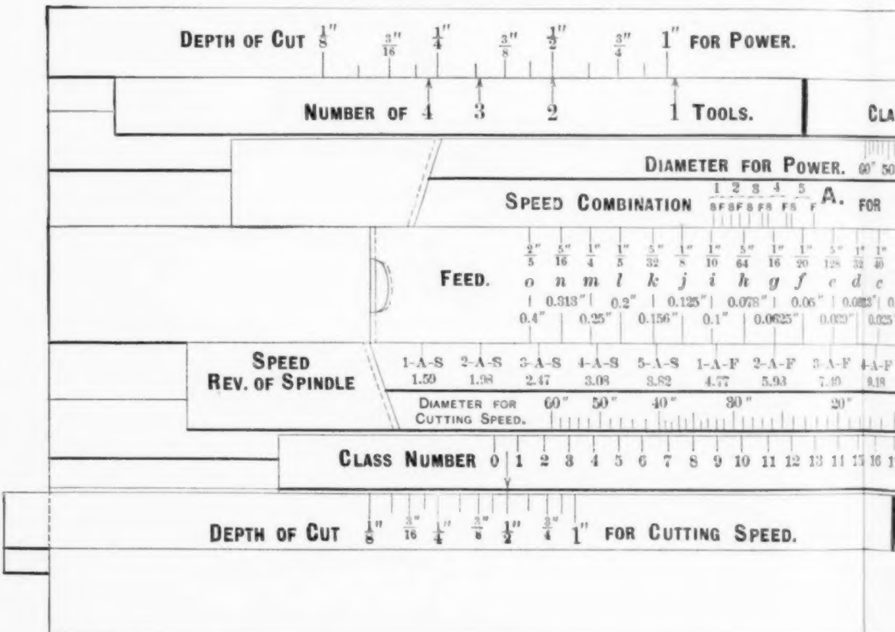


FIG. 4.



Barth, C.G.

FIG. 5.

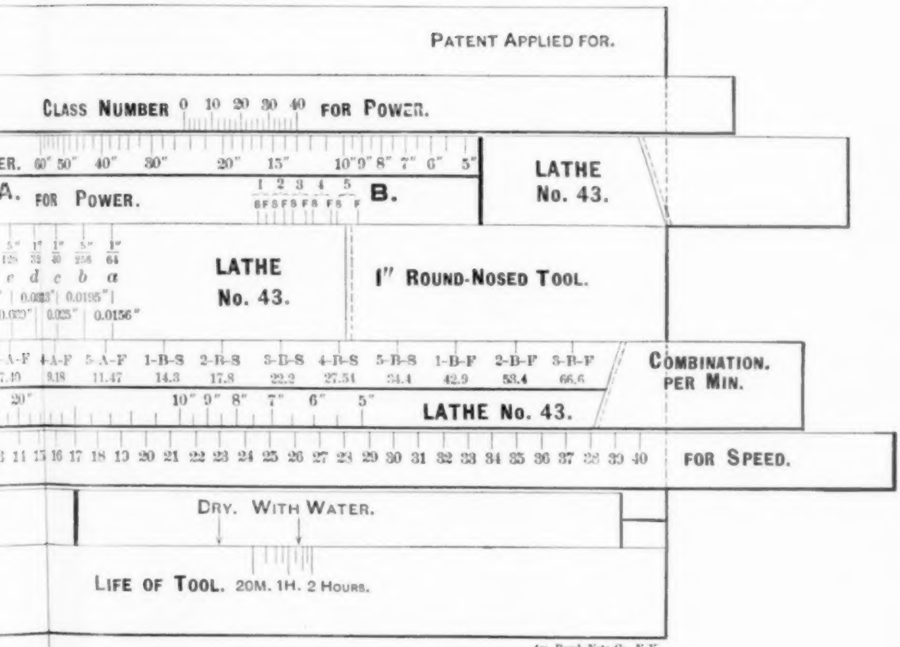

PATENT APPLIED FOR.
CLASS NUMBER 0 10 20 30 40 FOR POWER.
60" 50" 40" 30" 20" 15" 10" 9" 8" 7" 6" 5"
A. FOR POWER.
1 2 3 4 5
B.
LATHE No. 43.
LATHE No. 43.
1" ROUND-NOSED TOOL.
1-B-S 2-B-S 3-B-S 4-B-S 5-B-S 1-B-F 2-B-F 3-B-F
14.3 17.8 22.2 27.51 34.4 42.9 53.4 66.6
COMBINATION. PER MIN.
10" 9" 8" 7" 6" 5"
LATHE No. 43.
16 17 18 19 20 21 22 23 24 25 26 27 28 29 30 31 32 33 34 35 36 37 38 39 40
FOR SPEED.
DRY. WITH WATER.
LIFE OF TOOL. 20M. 1H. 2 HOURS.
Am. Bank Note Co., N.Y.

FIG. 5.

to it in the two x scales on the slides; and this done, we readily discover in which direction we must move along the first scale in order to pick out that value of y which has the same value of x coincident with it in both x scales. For the case under consideration this value of y is $7\frac{1}{2}$, and the coincident value in both scales is $4\frac{1}{2}$. Evidently, therefore, $y = 7\frac{1}{2}$ and $x = 4\frac{1}{2}$ are the numbers sought.

24. In the same manner we may make a slide rule for the solution of the general problem: "*The product and quotient of two numbers being given, what are the numbers?*"

Such a rule would differ from the above described rule merely in having logarithmic scales instead of plain arithmetic scales.

25. By the combined use of both arithmetical and logarithmetic scales we may even construct rules for a similar solution of the general problems: "*The sum and product, or the sum and quotient, or the difference and product, or the difference and quotient, of two numbers being given, what are the numbers?*" and a multiplicity of others; and the writer ventures to suggest that slide rules of this kind, and some even simpler ones, might be made excellent use of in teaching the first elements of algebra, as they would offer splendid opportunities for illustrating the rules for the operations with negative numbers, which are such a stumbling block to the average young student.

26. We now have sufficient idea of the mathematical principles involved, for a complete understanding of the working of the slide rule whose representation forms the main purpose of this paper.

27. This slide rule, in a somewhat ideal form in so far as it is made out for neither steel nor cast iron, but for an ideal metal of properties between these two, is illustrated in Fig. 5. It will be seen to have two slides in its *upper section* and three in its *lower section*, and it is in so far identical with the rules made for the Bethlehem Steel Company, while in the rules more recently made it has been found possible and convenient to construct it with only two slides in the lower section also.

28. It is shown arranged for a belt-driven lathe (No. 43*) with five cone steps, which are designated respectively by the numbers 1, 2, 3, 4, 5, from the largest to the smallest on the machine. This lathe has a back gear only, and the back gear in use is desig-

* The main frame of the rule is used for a number of lathes, and is arranged to receive interchangeable specific scales for any lathe wanted, as may be seen in the illustration.

nated by the letter *A*, the back gear out by the letter *B*. It also has two counter shaft speeds, designated respectively by *S* and *F*, such that *S* stands for the slower, *F* for the faster of these speeds.

29. The Speed Combination 3—*A*—*S* thus designates—to choose an example—the belt on the middle cone step, the back gear in, and the slow speed of the countershaft; and similarly, the combination 1—*B*—*F* designates the belt on the largest cone step on the machine, the back gear out, and the fast speed of the countershaft; and so on.

30. The double, fixed scale in the middle of the rule (marked Feed) is equivalent to the *y* scale of the rule in Fig. 4, and the scales nearest to this on the slides on each side of it (marked Speed Combination for Power, and for Speed, respectively) are equivalent to the *x* scales on the rule in Fig. 4. The rest of the scales represent the various other variables that enter into the problem of determining the proper feed and speed combination to be used, fixed values being either directly given or assigned to these other variables, in any particular case under consideration.

31. The upper section of the rule embodies all the variables that enter into the question of available *cutting pressure* at the tool, while the lower section embodies all the variables that enter into the question of *cutting speed*; or, in other words, the upper section deals with the *pulling power* of the lathe, the lower section with the *cutting properties* of the tool; and our aim is primarily to utilize, in every case, both of these to the fullest extent possible.

32. The example for which the rule has been set in the illustration is:

A ½ *inch* depth of cut to be taken with each of two tools on a material of class 14 for hardness, and of 20 *inches* diameter, and the tools to last 1 *hour and* 45 *minutes* under a good stream of water.

33. The steps taken in setting the rule were:

1. The first scale in the upper or Power section of the rule, from above, was first set so that 2 in the scale marked Number of Tools became coincident with ½ *inch* in the fixed scale marked Depth of Cut for Power.

2. The second slide in this section of the rule was so set that 20 *inches* in the scale marked Diameter of Work for Power became coincident with 14 in the scale marked Class Number for Power.

3. The first slide from below, in the lower or SPEED section of the rule, was so set that the arrow marked WITH WATER became coincident with 1 *hour* 45 *minutes* in the fixed scale marked LIFE OF TOOL.

4. The arrow on the lower side of the second slide in this section of the rule was set to coincide with $\frac{1}{2}$ *inch* in the scale marked DEPTH OF CUT FOR CUTTING SPEED.

5. The third and last slide in this section was so set that 20 *inches* in the scale marked DIAMETER OF WORK FOR CUTTING SPEED became coincident with 14 in the scale marked CLASS NUMBER FOR CUTTING SPEED.

Let us now separately direct our attention to each of the two sections of the rule.

34. In the POWER section we find that all the speed combinations marked *B* (back gear out) lie entirely beyond the scale of feeds, which means that the estimated effective pull of the cone belt reduced down to the diameter of the work, does not represent enough available cutting pressure at each of the tools to enable a depth of cut of $\frac{1}{2}$ inch to be taken with even the finest feed of the lathe. Turning, however, to the speed combinations marked *A* (back gear in), we find that with the least powerful of them (5—*A*—*F*) the *e* feed, which amounts to $\frac{5}{128}$ inch = 0.039 inch, may be taken; while the *f* feed, which amounts to $\frac{1}{20}$ inch = 0.05 inch, is a little too much for it, though it is within the power of the next combination (5—*A*—*S*), and so on until we finally find that the most powerful combination (1—*A*—*S*) is nearly capable of pulling the *i* feed, which amounts to $\frac{1}{10}$ inch = 0.1 inch.

35. In the SPEED section of the rule we likewise find that all the *B* combinations lie beyond the scale of feeds, while we find that the combination 5—*A*—*F* (which corresponds to a spindle speed of 11.47 revolutions per minute), can be used in connection with the finest feed (*a*) only, if we are to live up to the requirements set for the life of the tool; while the next combination (4—*A*—*F*) will allow of the *e* feed being taken, the combination 3—*A*—*F* of the *f* feed, and so on until we finally find that the combinations 3—*A*—*S* is but a little too fast for the coarsest (*o*) feed, and that both of the slowest combinations (1—*A*—*S* and 2—*A*—*S*) would permit of even coarser feeds being taken, so far as only the lasting qualities of the tools are concerned.

36. We thus see that there is a vast difference between what the

POWER section of the rule gives as possible combinations of feeds and speeds for the utilization of the full pulling power of the lathe, and what the SPEED sections of the rule gives for such combinations for the utilization of the tools up to the full limit set. However, by again running down the scale of feeds we find that, in both sections of the rule, the *i* feed ($\frac{1}{10}$ inch = 0.1 inch), is but a trifle too coarse for the combination 1—*A*—*F*, while the *h* feed ($\frac{5}{64}$ inch = 0.078 inch) is somewhat too fine in connection with this speed combination 1—*A*—*F*, both for the full utilization of the pulling power of the belt on the one hand, and for the full utilization of the cutting efficiency of the tools on the other hand.

37. In this case, accordingly, the rule does not leave a shadow of doubt as to which speed combination should be used, while it leaves us to choose between two feeds, the finer of which does not allow us to work up to the full limit of either the belt or the tools, and the coarser of which will both overload the belt a trifle and ruin the tools a trifle sooner than we first intended to have them give out.

38. The final choice becomes a question of judgment on the part of the *Slide Rule and Instruction Card Man*, and will depend upon how sure he is of having assigned the correct CLASS NUMBER to the material or not; and this latter consideration opens up a number of questions in regard to the practical utilization of the rule, which for the lack of time cannot be taken up in the body of this paper, but which will be fully answered by the writer in any discussion on the subject that may arise.

39. Having decided upon the speed and feed to use, the Instruction Card Man now turns to the TIME slide rule illustrated in Fig. 6, and by means of this determines the time it will take the tools to traverse the work to the extent wanted, and making a fair allowance for the additional time consumed in setting the tools and calipering the work, he puts this down on the instruction card as the time the operation should take.

40. For finishing work the pulling power cuts no figure, so that this resolves itself into a question of feed and speed only; and for the selection of the speed combination that on any particular lathe will give the nearest to a desired cutting speed, the SPEED slide rule * illustrated in Fig. 7 is used.

41. It will readily be realized that a great deal of preliminary

* Described in the *American Machinist* of November 20, 1902.

work has to be done before a lathe or other machine tool can be successfully put on a slide rule of the kind described above. The feeds and speeds and pulling power must be studied and tabulated for handy reference, and the driving belts must not be allowed

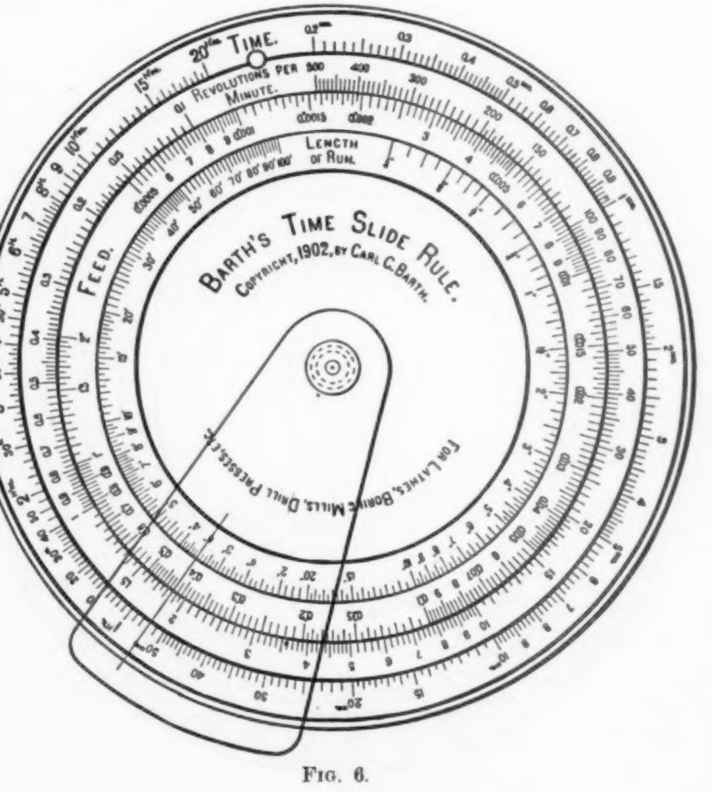


FIG. 6.

to fall below a certain tension, and must, in every way, be kept in first-class condition.

42. In some cases it also becomes necessary to limit the work to be done, not by the pull that the belt can be counted on to exert, but by the strength of the gears, and in order to quickly figure this matter over the writer also designed the GEAR slide rule * illustrated in Fig. 8, which is an incorporation of the formulæ established several years ago by Mr. Wilfred Lewis.

* Described in the *American Machinist* of July 31, 1902.

43. For the pulling power of a belt at different speeds, the writer has established new formulæ, which take account of the increasing sum of the tensions in the two sides of a belt with increasing effective pull, and which at the same time are based on the tensions recommended by Mr. Taylor in his paper entitled "Notes

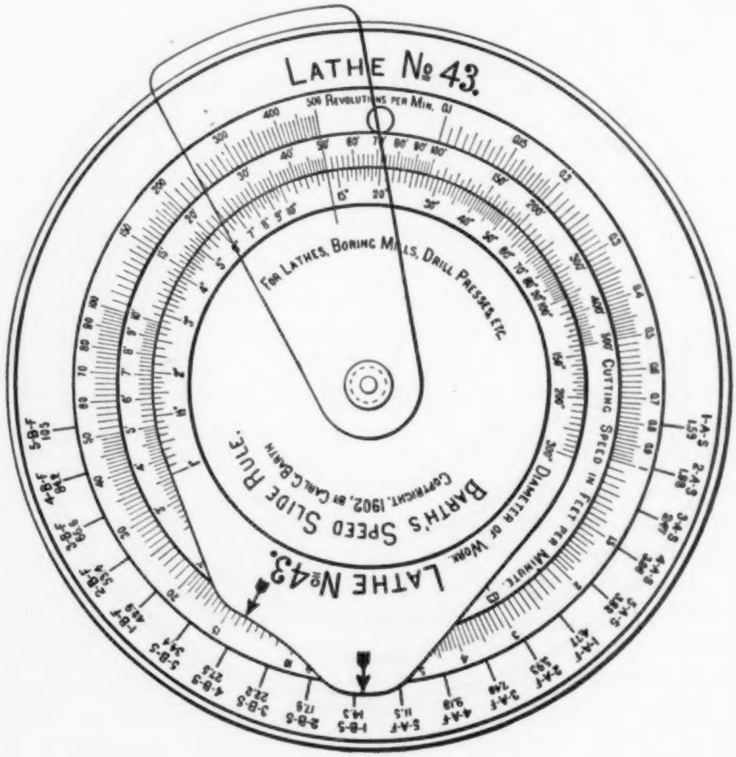


FIG. 7.

on Belting," which was presented at the Meeting of the Society in December, 1893.

44. These formulæ have also been incorporated on a slide rule, but as the writer hopes at some future time to prepare a separate paper on this subject, he will not go into this matter any further at the present time.

45. Having thus given an outline of the use of the slide rule system of predetermining the feeds and speeds, etc., at which a machine tool ought to be run to do a piece of work in the shortest

possible time, the writer, who has made this matter an almost exclusive study during the last four years, and who is at present engaged in introducing the Instruction Card and Functional Foremanship System into two well-known Philadelphia machine shops, which do a great variety of work in both steel and cast iron, will

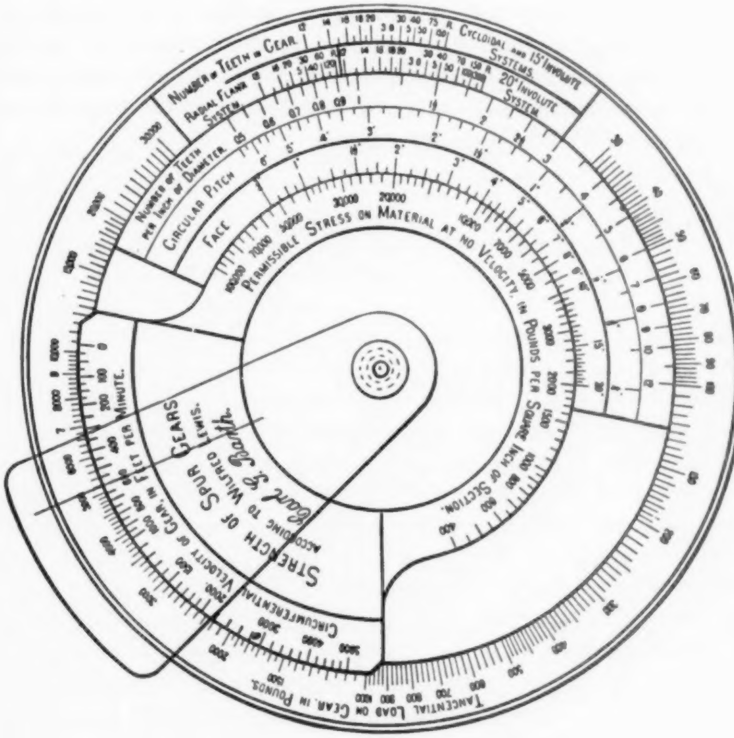


FIG. 8.

merely add that, in view of the results he has already obtained, in connection with the results obtained at Bethlehem, the usual way of running a machine shop appears little less than absurd.

46. Thus already during the first three weeks of the application of the slide rules to two lathes, the one a 27 inch, the other a 24 inch, in the larger of these shops, the output of these was increased to such an extent that they quite unexpectedly ran out of work on two different occasions, the consequence being that the superintendent, who had previously worried a good deal about how to

get the great amount of work on hand for these lathes out of the way, suddenly found himself confronted with a real difficulty in keeping them supplied with work. But while the truth of this statement may appear quite incredible to a great many persons, to the writer himself, familiar and impressed as he has become with the great intricacy involved in the problem of determining the most economical way of running a machine tool, the application of a rigid mathematical solution to this problem as against the leaving it to the so-called practical judgment and experience of the operator, can not otherwise result than in the exposure of the perfect folly of the latter method.

No. 1011.*

MODIFYING SYSTEMS OF MANAGEMENT.

BY H. L. GANTT, SCHENECTADY, N. Y.
(Member of the Society.)

1. At the Saratoga meeting the papers on shop management and the allied subjects covered such a broad field that a thorough discussion of them in the time available was practically impossible, and consequently, as the writer has since found in going over the subject with members, many of the most important points in those papers were not brought out.

2. Most of the people the writer has talked with regarded the various things advocated as individual propositions, and approved or condemned according as they saw, or did not see how they could, or could not be adapted to their works and their existing system of management. Many people apparently would like to adopt some of the ideas, but do not see how they can do so without making radical changes. This is due to the fact that few plants have in practical operation the basis on which the whole system depends for its ultimate success, namely a complete and accurate system of getting a record of all work done each day, and the amount of time spent in doing it. In other words a means of knowing in the office whether every order has been properly carried out or not, without actually going out into the shops and investigating. Whether the whole system as advocated is adopted or not this part is of great value to any manager, superintendent or foreman, and can usually be gradually introduced without stirring up any serious opposition. As a matter of fact it has been the writer's experience that all good men welcome such a system and do all they can to get it in operation; and, if a little time is taken, a radical change may be made by methods that are not at all revolutionary.

* Presented at the New York meeting (December, 1903) of the American Society of Mechanical Engineers, and forming part of Volume XXV. of the *Transactions*.

3. *A Complete System:* Referring again, however, to the views of the members regarding the methods advocated at the Saratoga meeting, it seems that few have grasped the idea that what was advocated was not a series of isolated propositions, but a system of management having a number of parts working in harmony with each other, designed first to find out in detail what the maximum output of a plant should be, and then to make it to the interest of all concerned to obtain day after day that maximum output.

4. *Different Views:* As an example of how diverse the criticisms of the papers were, I may quote some of the extreme ones.

5. One man, the manager of a large plant, thought that if the methods advocated by Mr. Taylor were attempted in his plant there would be a strike at once.

6. A second man did not see why we did not compel a workman to give the maximum output without extra compensation if we knew how it should be done.

7. A third man said to the writer, "Now for the first time I see how you can get the maximum output from your shop without having trouble with your men, and if you carry out the methods described I don't see how you can have trouble."

These opinions, being those of well-known and prominent men, are worthy of careful consideration.

8. *Analysis of Opinions:* Regarding the first opinion, that the introduction of the methods advocated by Mr. Taylor, and it was the study of unit times that was especially referred to, would produce a strike in his works, the reply is that *such study has never yet caused a strike.* That it is possible to cause a strike in almost any plant by doing this work in an obnoxious way is undoubtedly true; but it is equally true that the work may be done without serious opposition in almost any plant *if sufficient time and patience are devoted to it.* It is realized, however, only by those who have actually done the work how much time and patience may be needed, and a man who undertakes this work without the experience of others to guide him is apt to be discouraged, for he will find that his progress is extremely slow.

9. On the other hand if the workmen are so united in their determination that no one man can be found who will follow the wishes of the management and do exactly what is wanted without objecting to having the details observed and recorded, it would seem desirable to get such a man as soon as possible. This, how-

ever, is a condition the writer has never come across; and, while it may exist, is certainly very rare, and probably does not exist at all in any large plant, although it may take some time and patience to find the right man. When a start has been made and the good men begin to realize that what is being done is for their advantage as well as for that of the company, there need be no trouble provided men are allowed to see the results of one step before another is taken.

10. *The Second Opinion:* That if we knew how to get the maximum output of a machine we should compel the workmen to get it without extra compensation is not in keeping with the spirit of the age. We cannot to-day in this country compel anybody to do anything. The employer must concede to the workman what he demands for himself—that he be allowed to do what he believes it to be his interest to do. In other words, when the employer has decided upon what he believes to be his interest, his only successful method of procedure is to make it to the interest of the workman to do what is wanted. It may take time and patience to make the workman realize what his true interest is and see it in the same light as the employer does, and unions may oppose it; but if the inducement is a fair one and the employé is subjected to no real hardship it is only a question of time when somebody will be found of sufficient independence to work for his own interest.

11. *The Third Opinion:* That these methods properly carried out should give the maximum output and be practically an insurance against labor trouble the writer leaves for the comment of others. Suffice it to say that the writer is finding an increasing number of men that are looking at the matter in that light, and asking for information as to the best method of beginning this work under the conditions that exist in their shops.

12. *Introducing New Methods:* It must always be borne in mind that everybody is suspicious of new methods, and that the only way to remove this suspicion is to show them that the new methods are going to help them in their work. If the first thing that is started is helpful to somebody, it will not be long before there is a sentiment in favor of the new system. The time-keeping department and the foremen usually find the system of daily returns from the men of such assistance to them that it soon has their support, and when the Graphical Daily Balance begins to show up weak spots the best foremen realize they have at their

command an instrument which will help them to increase the efficiency of their work by enabling them to put their efforts where they are most needed.

Having thus stimulated an interest in making improvement, the value of the detail methods as advocated by Mr. Taylor will soon be realized, after which their adoption is only a question of time.

13. *The Office:* The remarks so far have been with reference to the shop, but they are equally applicable to the office, for to have a schedule of what should be done in the office each day and a graphical representation on that schedule of what was done is of great advantage to the management, and is essential to proper harmonious relations between the office and the shop. Indeed to be able to make quickly each day such comparisons as the following for the day before is quite as important as to make similar comparisons of the shop work:

What drawings should have been completed,
What drawings were completed,
What purchase orders should have been placed,
What purchase orders were placed,
What material should have been received,
What material was received.

14. It also costs but little to make readily available each day a knowledge of what has been spent in labor and material on any piece of work up to the close of the day previous.

DISCUSSION.

Mr. Chas. D. Parker.—The discussion has been almost wholly confined to the management and the workmen. I would like to make an inquiry of Mr. Gantt and others who have put the matter in practice: How about the intermediary agent, the foreman? What does he get out of it, and how much interested is he in putting it through? Does he get his pay raised when the men under him increase their earnings? Does he like it when some of the men under him make much more than he does, and is there any difficulty arising in putting this method in practice from that source?

*Mr. Gantt.**—I will say that the method adopted at the Beth-

* Author's Closure under the Rules.

lehem Steel Company's Works, when a number of men working under a foreman each had instructions for doing his work and a bonus for the accomplishment of that work in the time stated was as follows: the foreman was given a bonus in proportion to the number of men who earned their bonuses and an extra bonus if all of them earned it. It was thus made to the interest of the foreman to give his attention to the poorer workmen, and not to the best workmen, who needed attention least; that worked very satisfactorily at Bethlehem. I also know of another concern that is putting the same plan into operation now—an entirely different kind of plant.

No. 1012.*

IS ANYTHING THE MATTER WITH PIECE WORK? †

BY FRANK RICHARDS, NEW YORK CITY.
(Member of the Society.)

1. The title of this paper, if perhaps slangy in form is not so in fact, and is, I think, fairly characteristic of what follows. It may be frankly stated that the purpose is not so much to convey information as it is to provoke discussion and to accumulate knowledge upon one of the unsettled questions and one of the most important which can engage the attention of this Society. It is also one which can most affect the interests of its members and of those most interested with them in the safe and successful conduct of business. A perfunctory "sitting down" upon the paper is not all that any one could properly desire and cannot possibly close the case.

2. Attention is invited to the accompanying diagram (Fig. 9), which is easily understood. The purpose of it is to show the actual earnings of the workman, and of course also the labor-cost to the employer, for any given amount of work done under either day work or piece work at different rates, the Rowan premium system and Mr. Halsey's premium plan. The amount of work done is

---

* Presented at the New York meeting (December, 1903) of the American Society of Mechanical Engineers, and forming part of Volume XXV. of the *Transactions*.

† For further discussion on this topic consult *Transactions* as follows:

No. 341, vol. x., p. 600: "Gains Sharing" H. R. Towne.

No. 449, vol. xii., p. 755: "Premium Plan." F. A. Halsey.

No. 647, vol. xvi., p. 856: "Piece Rate System." F. W. Taylor.

No. 909, vol. xvii., p. 1040: "Drawing Room and Shop System." F. O. Ball.

No. 928, vol. xxiii., p. 341: "Bonus System for Rewarding Labor." H. L. Gantt.

No. 965, vol. xxiv., p. 250: "Gift Proposition for Paying Workmen." Frank Richards.

No. 1001, vol. xxiv., p. 1302: "The Machine Shop Problem." Charles Day.

No. 1002, vol. xxiv., p. 1322: "Graphical Daily Balance in Manufacture." H. L. Gantt.

No. 1003, vol. xxiv., p. 1337: "Shop Management." F. W. Taylor.

represented by the lengths of the horizontal lines and the wages paid are represented by the vertical lines.

3. As the Rowan system is not in use in this country all may not understand its basis of computation. It starts with a fair day's work, although that may not be the term used to designate it. The unit assumed is the amount or quantity of work which the man should ordinarily be expected to do in a day for the ordinary day's wage without any special inducement. The premium is earned only by the work which is done in excess of the regular day's work, and the premium earned is according to the time saved in doing the work. If double the work is done in the given time then one-half the time is saved and the man is paid one-half in addition to his regular wages. If the man does one and a half times his day's work then one-third of the time is saved and he is paid one-third more than his day's wages, and so on. The basis of computation is thus fixed and cannot be juggled with, but the inducement constantly decreases with the amount of work done, so that whatever a man may do he can never by any possibility double his earnings. Mr. Halsey's premium plan, of course, requires no explanation here, and it will be designated hereafter as *the* premium plan.

4. Referring to the diagram it will be seen that both day work and piece work, whatever the rate of the latter, are represented throughout by straight lines. A discouragement curve represents the Rowan premium system and Mr. Halsey's premium plan has a bend sinister. It was impossible to include Mr. Gantt's bonus system in the diagram because a part of it, the part where you do not quite earn the bonus, must be represented by an *invisible* line.

5. It cannot fail to strike the observer at once that in the premium plan the work which is done in the earning of the premium is straight, absolute piece work. The name cannot disguise it. The line in the diagram for the premium plan at one-half rate is exactly parallel to the half rate piece work line, the wages earned rise equally in each with equal increments of work done. So the three-eighth premium rate is parallel to the three-eighth piece work rate, and so on. If in making the premium plan bargain, the proposition were made to the man to first do his allotted quota and be credited with his day's wages and that then he should go to work by the piece for the remainder of the day at one-half the day rate, that would be the premium plan in every particular.

6. The partial piece-work character of the premium plan being undeniable, a paper whose topic is piece work must claim the right to handle it freely and without apology. The premium plan was invented by its originator nineteen years ago; it was put in operation in the shop at Sherbrooke, Canada, thirteen years ago, and was first brought to the notice of this Society in a paper twelve

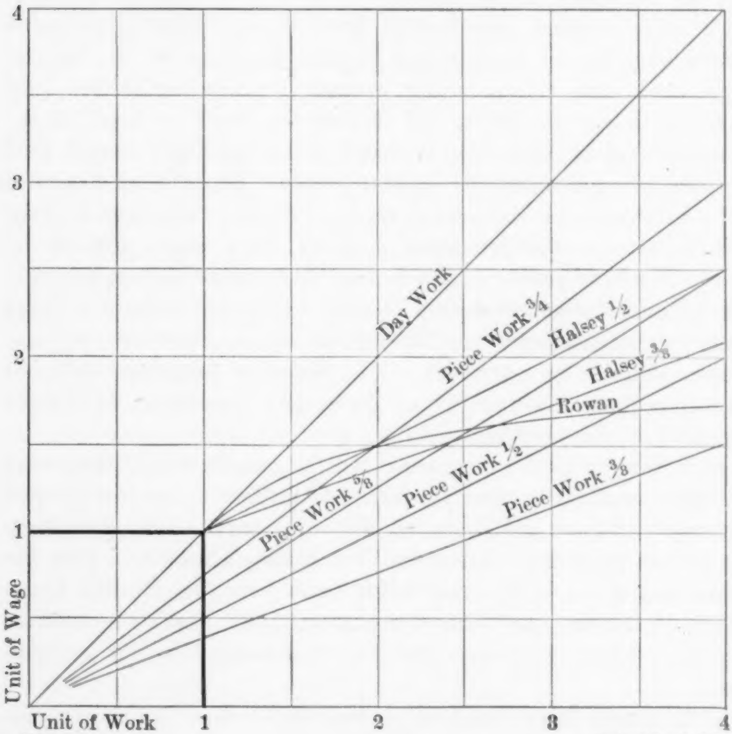


Fig. 9.

years ago. The plan, I know, has been proposed and advocated in all honesty of purpose; it has been pushed with earnestness and persistency. As a result the premium plan is in operation in a few machine shops and nowhere else. I venture the personal opinion, based on the fullest available information, that perhaps two per cent. of the machine work in the United States is done under the premium plan, while ten times as much is done by undisguised piece work and much more than half is still done by the day.

7. It is not at all apparent that there are any peculiar conditions in the machine shop which demand any different plans of wage adjustment than are prevalent in the other trades. While a knowledge of the premium plan is now widespread, the plan has not made itself appear so good a thing that any of the other trades have taken it up. It would not work with the shoemakers of Lynn, the hatters of Danbury, the glovemakers of Gloversville, or the stitchers and starchers in the collar shops of Troy, for they all work by the piece, as do most of the manufacturing trades, and the ultimate possibilities of economical production are thereby secured as completely as they can ever be claimed to be under the premium plan.

8. We might by an effort imagine the effect of proposing the premium plan to one of the trades outside the machine shop. Let it be tried on a lot of bricklayers. Say that it is first agreed that the day's wages are earned when five hundred bricks are laid, and that the premium plan begins right there. The bald proposition is, first, that if five hundred bricks are laid five hundred bricks will be paid for. This is so far meant to be an honest bargain on both sides. If you don't lay another brick above the five hundred we will have no cause of complaint. Well, now, having agreed to pay for the laying of the five hundred bricks, when the five hundred bricks are laid go on and lay as many more as you can. If you lay seven hundred and fifty bricks we will pay you for laying six hundred and twenty-five bricks; if you lay one thousand bricks we will pay you for laying seven hundred and fifty bricks, and so on. It will be very plain that under this arrangement the workmen are clearly the gainers, for if you lay more bricks you get *some* more money, and every additional cent you get is, of course, clear gain to you. The absurdity of this thing, when dealing with bricklayers, is sufficiently evident; are machinists so vastly different from bricklayers?

9. They must be different or else there are some things about the premium plan upon which I need information, and I take this way to get it. One of the inherent and inseparable conditions of the scheme would seem to be the voluntary acceptance of it by the individual workman. It depends entirely upon himself how much the man shall do after the allotted amount for the day's work is done. He may do much or he may do little, and therefore if he so chooses he may do none at all, but just be content to work along at his usual rate and just earn his day's wages. The

premium plan as I understand it is ostensibly, entirely a coaxing and not at all a driving plan; and yet it is a matter of common knowledge that in the State of New York alone there have been two determined strikes against the premium plan in the past year. This seems odd. If you don't choose to do what you are formally and distinctly allowed to choose whether you will do or not, what possibility for a strike can there be in that? Can it be that premium plan enthusiasts sometimes venture to put on to the plan some features which do not belong to it? I cannot imagine any other way in which a strike could be possible.

10. If they can tag things on and objectionably modify the premium plan they can also knock things off. The one essential safeguard of the premium plan continually insisted upon is that there shall be no cutting of rates when once established. This must inevitably involve injustice, because prices both of labor and of finished products change continually, and there must be, if justice is to prevail, sometimes a cutting of rates and sometimes an advance of rates. So far as it is possible to fix honest prices, and to maintain them there as long as it is just to both sides to do so, it can be done as well with straight piece work as with any premium plan, and is so done. For instance, I have knowledge of an establishment in the machine line, whose identity I must not disclose, where fifteen hundred men are employed and where piece work prevails in all departments, so that 90 per cent. of the productive work of the entre establishment is done by piece work, and it may be said of that establishment that there is no cutting of rates there, just as truly as I suppose it is ever said of works where the premium plan is in use. All prices when made run for a year. They are not arbitrarily imposed by the employer or his representatives, but are the outcome of fair and free and friendly conference, and when changes of price are imperative they are adjusted again in the same way. The works are prosperous continually, and the relation of employers and employés are less strained than they were under other arrangements.

11. It must be evident that none of these premium or bonus, or other curved or bent, or defective line schemes, whatever they may claim in the way of quickening the pace of the worker and increasing the output, can be the most effective, for the reason that they offer a reduced incentive at the precise time when the need of incentive is most urgent. It is the last piece done which comes the hardest, and it is absurd to offer the man half-price or

less for doing it. With either of the premium plans doing its best in the way of increased output and reduced labor cost per unit, and with piece-work prices adjusted to precisely the same price per piece, the inducement to the worker to increase his output still further must be greater under the piece work than under the premium plan. The guarantee that prices shall not be cut is precisely as applicable to piece work as to the premium plan. The latter has absolutely no monopoly of honesty, no assurance of price maintenance any more than the other. With equal temptation to cut, and with the same human nature in the boss, the chances of cutting will average precisely equal.

12. With no one having the slightest interest in pushing or advertising piece work, it is advancing on its merits as the most honest way of paying for repetitive work in the machine trade as in all others. It is worth while to note its popularity and progress especially in the extensive line of railroad work. The testimony at the meetings of the various railroad organizations is very pronounced in this direction. At the meeting this summer of the Railroad Master Blacksmiths one man stated that absolutely every job in his shop was done by the piece. When the price could not be placed on the work to be done it was placed on the "heat." Perhaps it may not always be possible to do this in the machine shop, but whenever the opportunity arises to consider the mode of payment it should always be in order to ask: Is Anything the Matter with Piece Work?

DISCUSSION.

Mr. Harrington Emerson.—Just two years ago I was fortunate enough to have Mr. F. W. Taylor explain to me his principles of shop directions, and the results that he had obtained. It is Mr. Taylor's great merit that he first applied these principles to machine shops, but they are as old as humanity, and consist in setting a definite task, in directing its execution and in apportioning the reward according to the deed. Thirty years ago I made my first acquaintance with these methods in the strictest of strict German schools, where we were put under functional teachers, ten or twelve different men each day, where we were allotted to the particular classes for which our uneven attainments fitted us: one boy in the highest English and lowest mathe-

matics, his twin brother perhaps in highest mathematics and medium English, where our methods, books, appliances, hours were strictly planned for us, our tasks set up to the full limit of our capacity, and where we were rewarded as individuals, not as members of a class. In the middle of the term a boy might be either promoted or debased, yet it was open to each to secure the same highest marks, and with the marks much-prized rewards and special privileges.

The results in that school were astonishing. The great majority of the boys learned more in eighteen months than other schools could teach in four years, because there was no waste time, no waste process, no waste effort; it was time unit study, functional foremanship and differential piece work.

At our meeting in Saratoga this year, Mr. Taylor presented in printed form a full statement of his fundamental principles, and, in my estimation, it will be many years before anything of more than detail value can be added to his work. My copy of his book is worn out with thumbing of the leaves, and marking of important passages. I discovered that it would have been easier to mark the few sentences here and there that were not of prime importance; yet, for our fellow-member, Mr. Frank Richards, it is as if Mr. Taylor's work had never been made public.

What is the matter with piece work? Everything is the matter with it. It is a lazy, haphazard method of shifting responsibility and direction from employer to employee. It works for deception in the latter, and gives us a long string of broken promises from the former; it is hated and opposed by the unions, and with reason; it brought on the great Union Pacific strike last year, which is not yet finally settled. Piece work makes no provision for justice, and any system is wrong that is not based on justice. In some of the great Burlington shops "Novo" and other modern steels have been introduced, doubling the output of the workman without extra effort on his part, yet the superintendent of motive power told me that piece work rates would not be changed, and in the Union Pacific shops men came and asked the privilege of paying for their own modern tools if piece rates would be left unaltered.

The fundamental trouble with piece work, in addition to its lack of justice, is that it makes the workman sell what is not his to sell, namely, OUTPUT. When Mr. J. J. Hill formulated

his famous principle that railroad expenses were by the train mile and receipts by the ton mile, neither his train crews nor himself ever dreamed of putting the pay of the men on a tonnage basis. The engineer who hauls sixty 80,000-pound cars with a hundred-ton engine gets no more than the engineer who obeys orders, standing for hours on a side track. The engineer sells his time, his skill, his intelligence, his obedience, but never output, because that depends on conditions over which he has no control; and it has always been a wonder to me that railroads which manage their train problems should be so backward in their machine-shop practices and methods.

What the employe sells, whether in office or shop, is not his "output," but primarily his time and his skill, incidentally his intelligence and his obedience.

That many shops pay by piece work is no argument in favor of the plan, since more shops pay by day work; and, as Mr. Barth in his slide rule paper, presented at this meeting, only too moderately remarks, the usual way of running a machine shop appears little less than absurd.

The experiences of Mr. Taylor, Mr. Gantt, Messrs. Dodge and Day, Mr. Barth, myself and Mr. Parkhurst, who have carefully studied the output and results in innumerable machine shops, prove that the wastes going on are more than absurd. As an example of old practice against new, I hold in my hands the original figures of the skilled and competent engineer of a large shop, who estimated the cost of a certain job at $4,575, of which $3,300 for materials and $1,275 for labor. The work came under my direction after it was one-third completed, and was pulled off with four men in three months for a total cost of $3,375.09, of which $622.79 for labor, netting a profit of $1,824.91, instead of $629, as estimated—nearly three for one, yet some of the men on that job were paid a bonus of nearly 100 per cent. above their regular wages. I also hold a routing card of one of my assistants, Mr. Parkhurst, in which a car shop job, marking and moving 200 pieces of oak, was estimated by the foreman to require two days, but was actually completed in 2 hours, 25 minutes on a 50 per cent. bonus basis. This is what Mr. Taylor's methods will do when applied to an old time shop.

In planning jobs of this kind we pay no attention to what has or is being done. Former practices have absolutely no interest for us. We figure out the time the job ought to take under

existing conditions, and we pay the man a generous bonus, which must be enough to call out the best that is in him. If the conditions appertaining to the job are changed, either for better or worse, we again determine the minimum times and pay the man a bonus for his coöperation. These illustrations show that astounding results follow the plans Mr. Richards condemns without understanding them, and that there is no argument whatever in appealing to present practices.

Mr. Richards's diagrams and his reasoning and conclusions are erroneous, because he bases them on output, which does not properly enter into the matter at all, as diagrams based on time instantly show.

I assume in all cases wages of 25 cents an hour, a usual time 8 hours for a given job, a slow time of 10 hours, a fair time of 6 hours, a piece work time of 5 hours, a Taylor bonus time of 3 hours. Time, days, hours, minutes—in this case hours—are measured on horizontal lines, wages by the week, day, hour or minute—in this case hour by the hour—are measured vertically.

DIAGRAM 1. DAY WORK.

(a) slow day, (b) average, (c) fast day under good foreman.

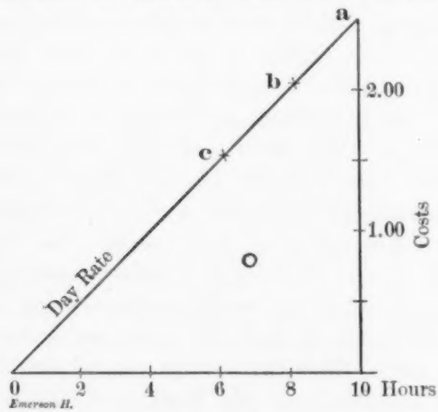


FIG. 10.

Normal cost,	8 hours	to employer,	$2.00 ;	wages per hour,	25 cents.
Slow	"	10	"	"	"	2.50 ;	"	"	"	25 "
Low	"	6	"	"	"	1.50;	"	"	"	25 "

The employer makes all the gain or loss. He is stimulated to good foremanship and better equipment, but the constant tendency is to deterioration.

DIAGRAM 2.—PIECE WORK.

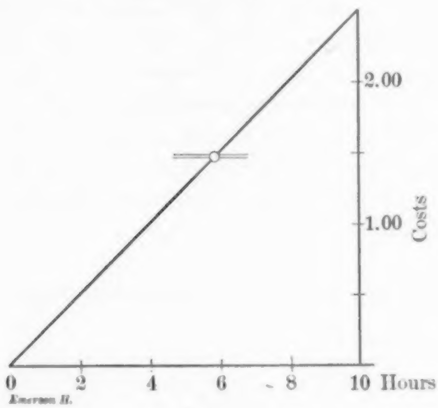


FIG. 11.

Piece work cost at 6 hours, $1.50 ; wages per hour, .25 cents.
" " " " 8 " 1.50 ; " " " .1875 cents.
" " " " 4 " 1.50 ; " " " .375 "

This is exactly the reverse of day work. The employe makes all the gain or loss, and is afraid to cut time for fear wages will be cut.

DIAGRAM 3.—HALSEY PREMIUM.

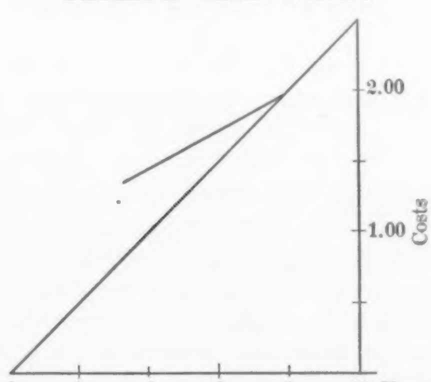


FIG. 12.

Cost to employer at 10 hours, $2.50 ; wages per hour, .25 cents.
" " " " 8 " 2.00 ; " " " .25 "
" " " " 6 " 1.75 ; " " " .29 "
" " " " 4 " 1.50 ; " " " .37.5 "

The chief merit of this plan is that it obviates the necessity for change in piece rates. It has worked admirably in certain shops, steering a half-way course between the injustice of day work and of piece work, but it is not fitted to cope with the unexpected. If there are no improvements by the employer there is no reason why the employe should not get in full the increased result due to his greater diligence and skill, but if improvement is due to the employer's better equipment there is no justice in giving the employe any part of it.

DIAGRAM 4.—TAYLOR DIFFERENTIAL PIECE.

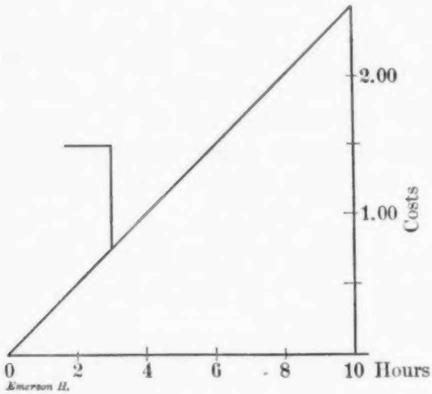


FIG. 13.

Cost to employer at 3 hours, $1.50; wages per hour, .50 cents.
" " " " 2 " 1.50; " " " .75 "

If employe habitually falls below three hours he is not wanted.

Here, for the first time, attention is concentrated on the reasonable maximum of production and the reward made proportionately great. Not only is there no attempt made to cut piece work prices, but the reward is withheld unless the maximum is done. The great difference between this and ordinary piece work is that Mr. Taylor demands the payment of a high premium, often 100 per cent., a figure that would frighten most employers, in order to effect maximum reduction in cost. If the employer introduces improvements, times are with justice shortened but not the premium per hour; if equipment deteriorates times must be lengthened but the same premium be paid per hour.

DIAGRAM 5.—GANTT BONUS.

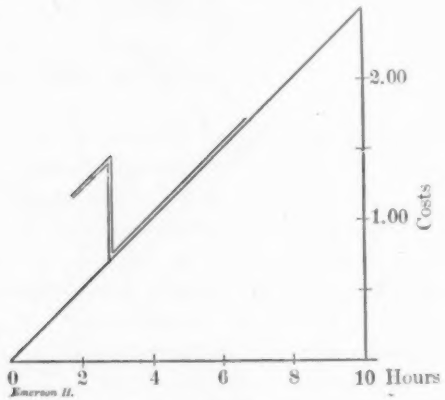


FIG. 14.

Cost to employer at 6 hours, $1.50 ; wages per hour, .25 cents.
" " " " 3 " 1.50 ; " " " .50 "
" " " " 2 " 1.25 ; " " " .625 "

The difference between the Taylor and Gantt plans is that the former pays by the piece finished in a definite time, while the latter pays by the definite time for a completed job, and pays the bonus, not for the piece, but for following instructions.

Mr. Gantt does not admit that under his system the workman could better the time set and therefore objects to the supposition of two hours on a three-hour job; but I extend the diagram theoretically in order to show the difference between Taylor and Gantt. Taylor is more severe and more generous.

After careful study of the Taylor and Gantt diagrams, Mr. Parkhurst and myself, adhering absolutely to the Taylor and Gantt theory of time unit study and specific directions of all operations, have used other diagrams, less severe than Taylor and Gantt, and permitting us in an old shop, where tool, machine and labor conditions are not modern, to keep the ideal always in view, yet we reward any gain shown by the workman. We determine with all the skill at our command the time a job should take, and adopt the Taylor line, based however on time and not on piece, and then run back to the day line.

DIAGRAM 6—EMERSON PARABOLIC.

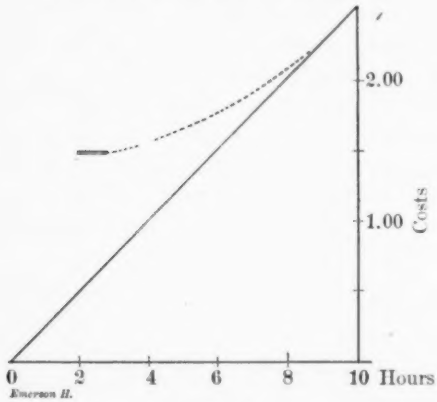


FIG. 15.

Cost to employer,	10	hours,	$2.50 ;	wages	per	hour,	.25	cents.
" " "	8	"	2.00 ;	"	"	"	.26	"
" " "	6	"	1.75 ;	"	"	"	.29	"
" " "	4	"	1.50 ;	"	"	"	.375	"
" " "	3	"	1.50 ;	"	"	"	.50	"
" " "	2	"	1.50 ;	"	"	"	.75	"

DIAGRAM 7—PARKHURST COMBINATION.

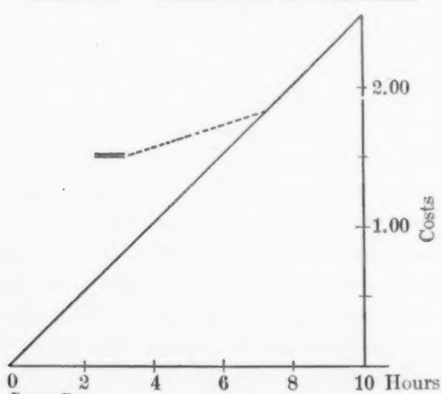


FIG. 16.

Cost to employer,	8	hours,	$2.00 ;	wages	per	hour,	.25	cents.
" " "	6	"	1.68 ;	"	"	"	.28	"
" " "	4	"	1.56 ;	"	"	"	.39	"
" " "	3	"	1.50 ;	"	"	"	.50	"
" " "	2	"	1.50 ;	"	"	"	.75	"

The essential difference between these diagrams and the Halsey premium line is not that they are curved and it is straight, but that it begins with an accurate and probably justly determined rate and drifts mathematically, but not scientifically, into space.

Mr. Parkhurst and myself begin with the scientific maximum of output and reward for endeavor, and, as a mere matter of shop convenience curve backwards to the day rate line. There is no special merit in the parabola or in the straight line; other lines might answer practically as well. The main point is that a little improvement gets a little taste of reward, and a big improvement gets a great big reward.

When all conditions are properly under control, I much prefer the Taylor diagram based on time. There is something inspiriting in working out a minimum time, in knowing that it can be made with the regularity that a train makes its fast schedule, in proving it, in stimulating the workman to it; but it is equally discouraging to workman, to expert and to employer to be wrecked in full flight by hard iron from the foundry, by variable speed in the engine, by broken belt on main shaft, by any unforeseen and unforeseeable delay, and in such cases the curve back to day rate prevents much trouble.

In all these diagrams, except day rate, the employe is benefitted by reduction in time; in all these diagrams, except piece work, the employer is benefitted by reduction in time, and reward for reduction in time is apportioned exactly as it should be, only by the Taylor method and its modifications.

The employer must pay big bonuses or he cannot get results. He can afford to pay big bonuses, for even if he gives all the gain in time to the employe he makes on increased efficiency of plant and diminished overcharges.

Where, in all these lines and curves, when based on time, is there any support for Mr. Richards's contention that they cannot be effective because, as he claims, they offer a reduced incentive at the precise time when the need of incentive is most urgent. Exactly the contrary is true. They all of them offer, just as piece work does, ever increasing pay, which, if pushed to the theoretical limit, would reduce costs to almost nothing, and give the employe an infinitely large sum per day.

Mr. R. G. Schneble.—In answer to the author's question I may add my experience, and say emphatically: " Nothing."

The troubles heretofore attributed to piece work are really not inherent to the system, but to those who used the system, they are the results of ignorance, carelessness and cupidity.

Improper rates are the result of ignorance or carelessness. When an operation or a series of operations are carefully analyzed by a person whose knowledge qualifies him, there is rarely any trouble after the employees are impressed with the fact that fair wages may be earned at the rate fixed. The difficulty heretofore has been that rates were carelessly made, and such a rate is nearly always high, and soon leads to restriction of output or excites the cupidity of his employers, the inevitable in either case being rate cutting and strikes.

It is much the same qualifications that keeps it out of at least one-half of the shops of this country. The cupidity of employers who have risen from small beginnings, who in their beginning were able to hold all the details in their own heads, add one foreman after another from the ranks, or from (which is worse) the family. It is impossible to recruit the necessary ability from such material. They absolutely refuse to pay for this ability, and all system or work done that cannot be written down on a customer's bill is wasted.

I see nothing in the Premium plan nor the Bonus plan, for if proper care is exercised in setting the time limits, a price might as well be set at once. 'Tis a sugar coating to fool the workman, or at best a stepping-stone toward piece work. In the Rowan system there is a corrective applied for excessive time limits (equals piece price). However, in the Rowan works the time limits are never changed (if I understand rightly), no matter if methods or machinery are improved.

I most heartily endorse the paper of Mr. Richards. I might add that in handling odd work in a piece-work shop I found it convenient to adopt what we termed an "excursion rate" good for this day and date only, until such time when the work became regular and it was possible to set a stable price.

Mr. Gantt.—With regard to Mr. Emerson's statement that I was not serious when I suggested that when a workman did better than his instructions called for he should be made instructor, I have to say that I was quite serious, and as a matter of fact one of the best instructors I know was discovered just that way.

Mr. Richards' suggestion with regard to the West Albany

shops is quite in order, and the superintendent of those shops is quite in accord with him; so much so indeed that he has strongly recommended that this work be begun there as soon as possible. The knowledge of what has been done in Schenectady is the reason for his action, which is seconded by some of his foremen who know the foremen at Schenectady.

Mr. Oberlin Smith.—There is one feature in all this discussion which needs to be gone into further. Everything that has been said seems to apply to manufacturing shops. Now we must differentiate between these and the most primitive kind of a shop, the jobbing shop, which is simply used for making odd things for people and for doing repairs. Therein we cannot well have anything much better than "day work." For this we must have the right kind of a foreman and the right kind of workmen, specialists in their line. Now, at the other end of the series, is a manufacturing shop where hundreds and thousands of things are all made exactly alike; or, at least, there are various component parts of them made alike. Therein systems of "piece" work or "premium" work or "bonus" pay are, of course, applicable, and they seem to have been thought out and developed especially with reference to such shops. But there is an intermediate kind of shop, and probably the great majority of our machine shops are of this class, whether you call it mongrel or hermaphrodite, or what not; semi-manufacturing would perhaps be a good name for it. It has chanced that my experience has been mostly in a shop of this kind. Our product is presses and dies for working bar and sheet metals. The dies are almost all different; they have to be worked on the jobbing shop plan in a department by themselves, and it would be very difficult to apply any "piece" or "premium" system to the work upon them. In our other department we make about five hundred different kinds and sizes of presses. In consequence of making so many kinds in a comparatively small shop, employing from 150 to 200 workmen, it has seemed very difficult to put in practice any of plans of payment in question.

Doubtless some of the gentlemen here have had experience in shops of this kind and have tested these improved methods of payment. The larger parts of our machines, such as the frames, and the component parts of such frames as are of built-up construction must, on the larger sizes, be made only one at a time, as some certain kind of machine may perhaps be ordered only

once in a year or two. Of course, very few of the pieces are made alike and cannot be made up in stock to much extent. Other more standard kinds are made up in batches of ten. We must, therefore, consider things which are made in batches of from one to ten on the larger pieces, and, we will say, from one to fifty on the small pieces. I have tried the premium plan and the piece work plan a little, but generally we had to fall back on the old day-work system on account of the great amount of bookkeeping involved.

When we try to manufacture strictly, we are handicapped by the small batches. Furthermore, we are much bothered when customers require modifications and insist that a casting must be altered; sometimes they merely want changes in the way of equipping with attachments often varying in design. Now, with all these complications how far can we apply any of the new systems?

In regard to the relative merits of the "piece work" and the "premium" plan, I do not feel competent to speak. Of course, we are all liable to meet the difficulties involved in the labor problem. The "walking delegate" has fixed upon the term "piece work" a big black mark. After he found something was being done to get around him on that, he put another black mark over the word "premium," and then "bonus" came under the ban. Labor, however, is gradually being educated in economics and will, I believe, become more sensible. We must help it to learn, and must sometimes meet it part way—remembering that there are two sides to every question.

Mr. E. P. Bates.—I think Mr. Emerson has left an impression that he would not care to have remain with us. As I understood him, where a price was made for piece work and the mechanic did the work quicker than was expected, the profit all went to the mechanic, and where he did it slower the loss went to the mechanic. My experience is that it may cost from fifty cents to two dollars a day to furnish this mechanic with tools and superintendence, including all the items which go to making the cost of operating a plant, and I think that the operator of the plant is quite as much interested as is the mechanic in regard to time, and that the loss or the gain comes to the owner of the plant fully as much as to the working man.

Mr. Emerson.—Of course, the gentleman understands that I did not read my paper in full; I abridged it very much, but if

you read it when published in the *Transactions* of the Society, you will find this statement in it:

"The employer must pay big bonuses, or he cannot get results. He can afford to pay big bonuses, for even if he gives all the gain in time to the employe he makes on increased efficiency of plant and diminished overcharges."

Mr. Taylor.—At the risk of being prosy, and always coming up with some old remark every time this subject is before the Society, I want to say again that I think there is no real quarrel between any of the systems of payment in common use. It is a curious phenomenon that there are certain men who seem to wish to attach their names to some one comparatively small and unimportant element in management. They appear to be unable to see anything else in the whole line of management except their one chosen element and attempt to convince themselves and every one else that this is the whole art of management. Apparently Mr. Richards looks upon piece work as the whole of this art. I have not heard that he has been able to see any good in any other element.

Now, I wish again to say what most of you have heard a number of times, but perhaps some have not yet heard it, that to my mind there is no quarrel between the various systems of paying men which are in common use. I think each of these systems has its proper place; at least four of them can properly be used in the same shop and at the same time, providing the shop is large enough. Every large machine shop in the country should have, I should say, not less than four systems of payment going on at once in order to do the work in the most economical manner. There cannot be any quarrel, then, between "day work" and "piece work" and Mr. Gantt's "bonus plan" and the "differential rate system" of piece work since, as I pointed out in my paper on shop management last spring, each plan has its own individual field of usefulness; and I feel convinced that each one has a field that it is impossible for either the other systems to entirely fill. And again, outside of these plans I feel that the Towne-Halsey plan has a large field of usefulness, Mr. Richards to the contrary notwithstanding.

But back of all systems of paying men and underneath them all, and of vastly more importance than any system of paying men, lies the true remedy for the fundamental difficulty between the managers and their workmen. The great difficulty that

presents itself is that in most cases neither the one nor the other know accurately how much of any given work a good man can and ought to do in a day. It is only in rare cases that either the managers or the men know what really constitutes a day's work. And what I feel absolutely sure of and wish again to emphasize is that the only proper solution of the wages question, both as to the system for paying men to be employed and the compensation to be paid, lies in a scientific study of how much each man can do and ought to do; what a really first-class man properly suited to his work can do if he wishes to and if he has the proper appliances. It is this study that is so much more important than the adoption of any one system of paying men, that by comparison the differences between the latter sink into insignificance.

After the owner or manager is in possession of the exact knowledge of how long it ought to take to do the work, even the day work plan, which in many cases is perhaps the least satisfactory method, will produce much better results than any of the other systems without this knowledge. Of course, with the knowledge there is a choice, and each one of the four systems has its proper place in every shop.

Mr. Stephen W. Balkwill.—With regard to Mr. Smith's remarks about cost, it seems to me that he brought out some ideas that are worthy of consideration which come more into actual shop practice, except where the factory manufactures a specialty and makes nothing but repetition work which does not require any change of machines. Take, for instance, an article of which ten pieces would be required at one time and a thousand at another time. It is quite obvious that the machine to be used takes a definite amount of time to alter, for it is supposed that each of the ten pieces must be as accurately made as any of the thousand, and the machine must therefore be as accurately adjusted in either case and therefore makes the cost of production per piece of the fewer number of pieces a great deal higher than that of the greater number on account of having the same common fixed expenses.

Regarding Mr. Richards's remarks about the young man whom he referred to as boring tires, would say that this is a remarkable performance presuming it is on a single machine and paid for by the day which does not seem to leave much room for improvement on the piece work basis. Referring to the statement about the scarcity of good foremen and the best manner of getting

good results in that direction, I agree with the gentleman that as a general proposition they are not well enough paid for what is required of them, and if sufficient inducement is held out a good foreman can be had. I know of a particular case where a foreman has a sub-foreman under him who has charge of a piece work department, and although not as intelligent, yet he is able to earn more money than his foreman, which does not seem equitable nor liable to contribute to the good feeling of the foreman.

Mr. E. F. DuBrul.—I wish to dwell a little on a thought brought forward by Mr. Oberlin Smith. In this discussion he has brought up the question of the opposition of the labor unions to "piece work" and other methods of "Shop Management." Papers on "Shop Management" very frequently state that no serious difficulty is encountered from the workmen when such systems are installed. I should like to know whether this is a general rule in foundries. I have heard of considerable opposition on the part of moulders to the introduction of any other system but straight "day work." I have not heard of many foundries running anything but straight "day work" or straight "piece work," and under both conditions of "day work" or "piece work," I am informed that there is a very widespread disposition on the part of moulders to limit production and hold down output.

Coming to the machine shops we find that the last Convention of the Machinists' Union adopted a resolution declaring for the abolition of anything excepting straight "go-as-you-please day work," with no tasks set, no premiums, bonuses, "piece work," or anything of the sort. How many here are familiar with that promulgation? If this fiat is to go into effect, we must certainly reckon with the Union as an element in the problem. The Anthracite Coal Strike Commission well says that: Trades unionism is becoming a matter of business and that employer who fails to reckon with it as an element of his business makes a serious mistake and one which he will have to correct sooner or later.

Perhaps business conditions next year will make inadvisable the proposed movement to abolish "piece work." Coming events casting their shadows before appear in the two strikes in New York City this year, mentioned by Mr. Richards, against premium work. I happen to be familiar with the cause of those two

strikes and I can assure Mr. Richards that "piece work," premium work and all other similar methods look alike to the unions in question, and that the strikes would have occurred against "piece work" just as much as premium work. Furthermore, it seems to me that the employers in question would have had more difficulty in filling their shops with non-union piece workers than they had to fill them with non-union premium workers, because even a non-union man does not like "piece work" over much.

I do not believe that there is so much the matter with "piece work" as there is with the men who are trying to establish it in their shops. The greatest difficulty is with the managers who "know not Jacob." I had occasion to deal with such a case not over two weeks ago. As you may all know, premium work is very largely and very firmly established in the shops in Cincinnati and by general consent of the Associated Manufacturers in that city the rule is that once a time limit is established it shall not be cut unless a change has been made in the methods of production. A certain shop had a new superintendent, one of the kind who "knew not Jacob," who had never operated a premium system before and who thought to make a showing for himself with his employers by cutting down a time limit on a very efficient workman, who did a sixty-hour job in eighteen hours. Those of you who are familiar with the premium or other similar system will not question the statement that such reductions of time are common on the part of workmen, and the great difference is very largely because the men setting the time limits, while they may have a guide as to how long a job used to take, have absolutely no way of knowing how long a job ought to take.

The matter coming to my attention, it became my duty to interview that superintendent and show him the error of his way. His cut of a time limit was in violation of the guarantee that had been made to all the machinists of Cincinnati by the Associated Manufacturers, and his cut would have wrecked the system, not only in his own shop, but in all the others. If his ideas had been carried out, the result would have been a rejuvenation of the Walking Delegate, with whom we have not been bothered for some years, and I am sorry to say that it took quite a while and much vigorous language, some of which was unfit for publication, to demonstrate to that superintendent the inadvisability and injustice of his proposed action.

"The matter with piece work" is in my judgment principally

the matter with the employers or their representatives, who, generally through ignorance as to how long a job usually takes and occasionally through downright "hoggishness," have brought "piece work" into a disfavor it does not deserve. Were we all better informed, and did we all put piece prices, time limits, task work and other such systems on a scientific and accurate basis, and once so based did we guarantee the men against cuts in prices unless general reductions in day rates were made, I believe that all hostile criticism would be forever disarmed.

In the meantime, I believe that we should all keep our eyes on the union end of the proposition of "Shop Management," and not stumble along blindly in that regard any more than we should in regard to setting rates.

Mr. Smith.—I wanted to ask Mr. Taylor, when he spoke of its always being practicable to find out the cost of a certain job, whether he thinks it possible to get at such cost where only one machine is fitted up in a year and, if so, how he does it? Of course, if it is turning a steel shaft, it can be measured and the number of pounds to be taken off in chips can be estimated; but suppose it is an irregular casting of a new pattern, does it pay to find out, or can it be found out from the experience of only having one to make, what the cast will be on the next one?

Mr. Taylor.—I think the best answer to Mr. Smith's question will be found in Mr. Barth's paper, which will be presented to the Society at this meeting. Mr. Barth will show how an ordinary mechanic (with the aid of our slide rules) can determine accurately and quickly just what combination of cutting speed and feed should be used in any particular case in order to do the work in the quickest time on any given lathe planer or other machine tool, and in finding out the proper cutting speed and feed to use in cutting forgings or castings, each of the following elements which affect the answer is given its proper weight through its own slide on the slide rule. The variable elements, each of which affects the answer, are:

1. The pulling or driving power of the machine.
2. The strength of the feed mechanism of the machine.
3. The exact coarseness of the feed which can best be used.
4. The diameter of the piece to be turned.
5. The thickness of the layer of metal to be removed or the proper depth of cut to be taken.
6. The hardness of the metal which is being cut; *i. e.*, whether

cast iron, steel or brass, and the exact degree of hardness of the particular casting or forging.

7. The shape of the cutting tool and its size.

8. The quality of the tool steel from which the tool is made, and the heat treatment which the tool has received.

9. Whether water, oil, air blast or other cooling medium is used on the tool.

It is evident that this is a problem which is exceedingly difficult to solve. And the fact that these difficult problems are now being daily and most practically and rapidly solved by the ordinary mechanics who are using slide rules makes it evident that it is a comparatively simple matter to determine the time required to do the remainder of the work of running a machine, namely, putting the work into the machine, taking it out and adjusting the machine, etc. And the study of the time required to do this hand work is greatly simplified by dividing each job into its simple elements and then timing each element separately and systematically with a stop watch. Such, for instance, as:

Lifting work from floor to machine.
Putting on carrier.
Adjusting work in chuck or on centres.
Calipering.
Setting tool, etc.

If the "hand work" is studied by single simple operations, in this way it will be found that the entire hand work of a shop can be resolved into comparatively few elements which can be classified, tabulated and readily used in determining the proper time for doing even new and complicated work.

Mr. Smith.—Does that include the rigging and the unrigging of the tools?

Mr. Taylor.—Yes; it takes up the work from the time it leaves the floor to go into the machine until it comes out finished, and it includes the time required to make all of the changes and adjustments in the machines. The observations are stop-watch observations, carefully taken by a man who is trained to this business. Up to this time these observations have been recorded and tabulated on loose sheets or in manuscript books. But I predict that there will in the future be many books printed covering in each trade the time required to do each of the elementary operations which together in various combinations make up the entire work of the trade.

*Mr. John Calder.**—In Mr. Richards' paper too much is made of the identification of modern management advances with particular men and methods, and he appears thereby to miss the point.

I concur most heartily with Mr. Emerson in his praise of Mr. F. W. Taylor's paper of June last, which I believe will bear its full fruit until the appearance of the next volume of the Society's Proceedings.

To illustrate the point on which issue has been joined with Mr. Richards' I wish to present a practical problem in which straight piece work is maintained producing all that Mr. Richards claims for it, and a great deal more, through expense incurred by the management in scientific time study and its necessary accompaniments of betterment.

Where the Taylor system of management and labor reward has replaced straight piece jobs in manufacturing concerns doing nothing else at machine and bench but large quantities of light repetition work, we have an instance in which the issues between Mr. Richards and his critics can be reconciled. I believe that, in such circumstances, if the true time is scientifically determined after management and equipment conditions of the highest efficiency have been established, it does not matter in the least, whether (1) a large reward is offered for the daily task, (2) a piece rate is fixed to give the same maximum daily pay, (3) or a non-graduated premium or bonus appropriation which added to a fixed daily time rate will give the same reward.

Under such circumstances Mr. Richards' output and Mr. Emerson's time abcissæ would coincide, in fact, their diagrams would be identical.

In actual practice, workmen employed under above maximum output conditions can see no difference between shortening the time or increasing the task and lowering piece rates; and, as a matter of fact, there is none.

The one, but very important, improvement upon straight piece working being that accurate scientific time study has prescribed the maximum and fixed the rate for existing and improved conditions brought about on the initiative of the management, while ordinary piece rates are more liable to change, not because they are "piece" rates, but because they are more

* Contributed after adjournment.

or less guesses and depend for rectification purely upon what measure of their true skill the men choose to reveal.

For the class of work assumed, straight piece work based upon scientifically conducted time study and modernized facilities, will be as easy as any to establish in place of guess piece rates; and any other name given to the process is a distinction without a difference. It eliminates the inefficient, and provides an easy and automatic method of paying and encouraging the efficient but new men who take a little time to attain the maximum task—say coming at first within fifteen per cent. of it—while all others who aspire are tried out on daily time rates before graduating as "fit."

No. 1013.*

SUGGESTIONS FOR SHOP CONSTRUCTION.

BY F. A. SCHEFFLER.
(Member of the Society.)

1. Some time ago, the writer had occasion to lay out various buildings for one of the well-known electrical manufacturing firms in this country, with the view of constructing new shops, covering a complete equipment for manufacturing on a large scale.

2. At the time various schemes were suggested for the arrangement of the buildings in relation to each other, so that the resultant buildings could be combined into a scheme of interchange between the administration offices, sub-offices, and materials from one building to another.

3. After discussing the various ideas, above referred to, the form shown on the accompanying diagram, Fig. 17, suggested itself to the writer, and as it is entirely novel in its construction, so far as applied to machine shops and other manufacturing purposes, he deemed it advisable to place it on record, as there are many features connected with the layout, which although subject to modification, would make an ideal manufacturing shop in every way.

4. This plan was not carried out owing to the fact that the proposed new buildings were abandoned for several years, and when the works were eventually built other parties had the matter in charge, and the proposed plan was not even known to them.

5. Of course, it would have to be agreed upon in advance that there would be required for manufacturing purposes a number of buildings suitable for the various kinds of work to be manufactured. The plan herein proposed was primarily designed for manufacturing electrical apparatus, such as generators, motors,

* Presented at the New York meeting (December, 1903) of the American Society of Mechanical Engineers, and forming part of Volume XXV. of the *Transactions*.

switches, electrical instruments, etc.; but of course, the same scheme would be applicable to any other kind of manufacturing where it is desired to have a number of shops, all of which are easily accessible, both for business purposes and for delivery and shipping of material.

A brief description of the layout is as follows:—

In the center of the space available for the buildings is located

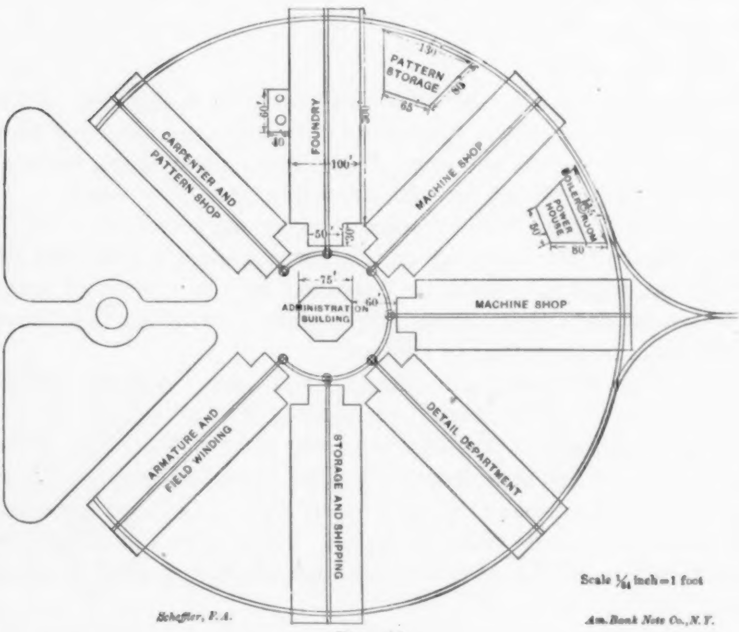


FIG. 17.

an administration building, constituting the business, accounting, and sales offices; and on the second story, the draughting room. This building is octagonal, or hexagonal, which ever may be found to be most suitable for the purpose. In this case, it has been designed with a view of accommodating seven buildings, which radiate from each side of the octagon, and has one side reserved for the main entrance through the building.

6. The end of each shop which is nearest to the administration building has its individual office for the foremen and shop clerks. This, it will be seen, is a very harmonious arrangement, as every shop is then but a short distance from the administration building,

so that intercourse can easily be had between the drawing room, offices and the offices of each shop.

7. The general arrangement gives each shop plenty of yard room, which is also very essential; and travelling cranes, either worked by hand or power, could be located in the yard room between any two of the shops, for handling raw or finished material.

8. A circular track around the administration building connected in front of each shop building by means of suitable turn tables, worked by hand or power, makes the distribution of material between the buildings very easy, and it will be noticed that the distance the material will have to travel from any one building to another is comparatively short.

9. At the extreme outer end of each building is another circular track, primarily to be used for shipping purposes, and the distributing of such material as may come in or go out over the connecting railroad lines. This track runs through the end of each building; and in such buildings where the machinery, castings, or other goods are to be handled, the heavier travelling crane in that particular building which should run the length of the shop, can easily unload or load the cars. This arrangement makes it possible to go into every shop without having a multiplicity of tracks and switches, thus cutting up the available yard room, as is usually the case in ordinary plants. It is also possible, if there is sufficient ground available, to extend any one or all of the buildings, and still retain the best features of the design.

10. A study of the design, which as above stated, is subject to modification, will be all that is necessary without any further comment, to make it clear to any one interested in this important question of the best arrangement of buildings for shop purposes.

11. In connection with this matter, I would add that while there is as far as the writer's knowledge goes, no manufacturing plant built on these lines, at the same time there is a plant of an entirely different character in Pennsylvania, where practically the same idea is carried out, so far as the location of the buildings, in relation to the office building is concerned. This is the Eastern Penitentiary, located in Pennsylvania, a cut of which was published in the *North American* recently, which the writer ran across by the merest accident, and it really makes a very good perspective picture of what a manufacturing plant would look like when laid out as above suggested, eliminating, of course, the walls surrounding the grounds.

DISCUSSION.

Mr. W. D. Ennis.—The arrangement of buildings illustrated by Mr. Scheffler possesses few advantages over that which is customary. Those which it does possess are incidental and accidental rather than dependent upon the eccentric distribution of departments proposed.

For example, the advantages of loading cars by cranes and of possible extension at low cost are possessed to an equal degree by this or almost any conceivable grouping of buildings.

The scheme is wasteful of land, requiring for the same yard and building area very much more ground space than is usual.

The diagram shows a centrally located office. It is questionable, however, whether it pays to sacrifice other considerations to those which in plants of any size private telephone systems and adequate messenger service seem generally able to satisfy.

It also shows one of the worst possible systems of trackage. There is but one communication with each department, which must be used both for ingoing and outgoing traffic. The trackage provision is not adapted to the requirements of the various departments. The same facilities are provided, for example, for the detail department as for the storage and shipping building. Should a carload of pig iron come in while a car of coal stood at the power house, extra switching would be necessary. All the trackage is in curves, which hampers the movements of cars and increases the probability of derailment and consequent delay. Access from the office to any of the buildings is impossible without crossing the industrial railway tracks, a condition which may counterbalance all the saving in time due to having the office centrally located. The presence of railroad tracks within the buildings has been found extremely dangerous to life and limb in grain elevators, and it would not seem advisable to introduce such an arrangement in manufacturing plants. Fire hazard would be increased by running locomotives through the departments, especially such as those housed in the armature, carpenter and storage buildings.

The power house is not centrally located, is of a wasteful and awkward shape and cannot be extended in any direction without heavy expense.

The yard space is accessible only from the rear; it is distributed without regard to departmental requirements, the detail depart-

ment, for instance, being given somewhat more room than the foundry. It is so shaped that it could not be properly covered by cranes, nor could any compact and systematic grouping of yard material be practised. As illustrated in the diagram, the space on the outside of the tracks is too far away from the buildings to make economical yardage, but unless it is utilized the full benefit of the track facilities will not be obtained.

It is difficult to see what advantage, excepting in the single point of accessibility from the office, the suggested arrangement of buildings has over that in which the departments are housed under separate roofs, side by side, with the office building on one side of the group and the railroad on the other side, one or more switches being run into the buildings where needed. The latter arrangement would certainly give better results as to amount and distribution of yard room, despatch and economy in shipping and receiving goods, and *convenient* access between office and shops and between the shops themselves. This arrangement might have to be modified to a moderate extent, but only in order to obtain direct access between successive departments, as, for example, the machine and finishing rooms of a paper mill. The feature of Mr. Scheffler's plan, requiring a journey around several corners to get from one building to another, should be absolutely prohibited in laying out any plant.

*Mr. Chas. L. Heisler.**—I offer the following criticism on the radial plan of arranging shop buildings:

1. There should be a material switch independent of the shipping switch. The plan shows that any car set for loading in the machine shop, must be reset and shifted each time a load of coke, coal or sand is delivered to the foundry, and the locomotive must each time pass through the machine shop. If another switch is made to parallel the one shown, then when the machine shop is extended there will be two switches passing through this department.

2. A curved switch is dangerous and seriously inconvenient to the switching crew, who cannot see two car lengths, so it is very difficult to set cars and avoid injuring shop men.

3. It is seriously objectionable to take a train of cars and locomotive through several shops for the purpose of reaching another.

4. The circular switch does not come within 150 feet of the

*** Submitted after adjournment.**

foundry cupola, and gives no opportunity for using hopper-bottom cars for cheaply handling foundry sand, coke, etc. The present arrangement will cause an expenditure of many hundred dollars per year in the extra handling of the raw foundry material.

5. The coal for the boilers should be dumped directly in front of the boilers. In the radial plan shown, each car must be unloaded by hand and reset every time any car is taken from either the foundry, pattern shop or machine shop.

6. The pattern storerooms should be adjacent to the pattern department, but sufficiently isolated for fire protection. The cupola should be farther from the pattern department, and should be located with respect to prevailing winds, if possible.

7. The arrangement does not comply with the present practice of arranging buildings as much as possible parallel to each other to economize in land and in order to utilize their crane columns and steel framing for supporting traversing yard cranes, which should cover all the available space adjacent to the several buildings, and which can be covered, when necessary, to meet future growth. The triangular yards evidently will never permit this without excessive expense and waste.

8. Assume that 30 per cent., more or less, covered floor space than shown on the plan is required for any one of the departments. First attempt to lengthen such a department 30 per cent., and note that the circular switch then divides the enlarged department, or, suppose the 30 per cent. enlargement consists of an unsymmetrical side addition, this cannot then be well fitted into the very undesirable form of triangular yards. On the other hand, assume that any department required 30 per cent. less floor space than shown, in this case the building would not come within 100 feet of the switch, or the central "administration" building.

9. The several receiving and shipping offices must be adjacent to the switch, and will, therefore, be 350 feet away from the main office, as shown.

10. The reduction in distances at other points between parallel buildings, when arranged as usual with an intervening crane yard, would certainly effect a greater saving than would be lost by the slightly greater distance between the main and other offices, as compared with the radial plan.

11. A moment's thought will make it clear that the alleged saving in time due to the radial plan is lost several times over in

the time required in manipulating the many heavy 12 x 14 foot switch doors. Assume that a switching engine and crew are making a trip around the circular track, the crew will be required to make at least 24 distinct and strenuous efforts in opening and closing the 12 heavy doors, and it will take 48 such operations in cold weather, if they are all lifting doors, and 96 if they are double doors. However cold the weather, one round trip, not considering the resetting of cars, I think would finish even the most robust switching crew. If they did survive one round, I fear the shop men would never permit them to make a second.

Mr. Suplee.—In reference to the statement made by the author of the paper just now about the arrangement of the buildings being the same in the Eastern Penitentiary at Philadelphia, I may add that in the centre of the circle formed by those buildings there is a system of mirrors so arranged that a single watchman seated in the centre can look down all of the corridors at once. This ingenious system of mirrors was designed by one of the prisoners.

No. 1014.*

WHAT ARE THE NEW MACHINE TOOLS TO BE?

BY JOHN E. SWEET, SYRACUSE, N. Y.
(Member of the Society.)

1. It is a fact quite apparent to users of machine tools that the new high-speed tool steel calls for a re-designing of our machines if we are to get even a fair share of the ultimate possibilities which the new steel offers.

2. I expect the machine tool builders have already the reply formulated as follows: "You just keep on building engines and leave the machine tool business to us." But that will not quite do. If no one but the engine builders had mixed in the engine business, we would have had no turbine engines, and many of the standard machine tools were originally devised by those who had use for them rather than by the man who devised things to sell.

3. I think the machine tool builders will admit that the machines must be re-designed; but to the most of them will this mean anything but just to make the driving elements more powerful and the machines stronger, which is as much as to say everything has been all right, and all we need to do is to change the strength and power. But have they been all right or half right?

4. It can be shown by figures, I suppose (I know it to be a fact by a trial with models), that a complete box is thirteen times more rigid against torsion and four times more rigid against bending than the same amount of material is in the form of side plates and thin cross girts. It is probably from four to eight times more rigid than the cross girt plan in any form, and yet in the case of lathes, the whole business of whose beds is to resist torsion, only one or two builders have had the courage to adopt the box form.

5. All planer beds can just as well be box beds with half the cost in patterns and foundry work, and so too the tables which are

* Presented at the New York meeting, December 1903, of the American Society of Mechanical Engineers, and forming part of Volume XXV. of the *Transactions*.

sprung by bolting down work can just as well be box tables four times as strong with the same material, and with a saving of half the cost in patterns and something in the foundry.

6. The whole tendency of the cut is to slide the work endwise of the planer bed; but who has ever tried putting the slots crosswise in a way to offer the greater resistance and prevent the bending of the bed by the peening of the upper surface, as now occurs, which with the springing by bolting down the work are the primary causes of cut ways.

7. Some planer and boring mill cross rails are of box section in the centre, but are thinned down at the ends when fastened to the housings. The most of them are three sides of a box only, or one-tenth the strength of a box, where a plain square box straight through is infinitely better and cheaper. Of course the boxes are not to be proportioned from what is in use now, but from what is to be made to meet the new conditions. To select enough material to meet the new demands and then put the material so that it will be four times more rigid will be something like it. Housings of box section will be just as rigid fore and aft and much more rigid against side strain.

8. Milling machines of the planer style are constructed like planing machines, seemingly without a thought but that the conditions are identical, while they are not. If the bed of a planing machine and the table were of the same length, the weight of the table and the load over-running the end of the bed would soon wear the top of the bed crowning and the under side of the table concave to fit, and it is to counteract this tendency of gravity to wear them out of true that the beds are made longer than the tables. With the milling machine the load is less, more of it in the middle of the table, because there is less gained by putting on small pieces end to end, and the down pressure of the big cutter always in the middle partially, if not wholly, neutralizes the tendency to wear out of true by gravity. When such a machine has side cutters or a vertical spindle, the pressure is always in the middle, first in one direction and then the other, exactly the reverse from the gravity action, and instead of the side guide of the bed being longer than the table it should be shorter, by just about the same amount as the bed of a planer needs to be longer.

9. Many times the sliding piece and its guides can be the same length and keep straight. The things which do not tend to wear out of true do not wear much, and the things which do wear out

of true and have to be refitted are never just right but when new and when so refitted. Where a short block slides on a long guide, if the scraper marks wear out sooner along the middle than at the ends, the ends of the guide need cutting off, however much over-run it gives to the sliding block.

10. The draughtsman dare not make a drawing of an engine cross-head over-running the guide one-third of its length at each end; the builder would hardly dare to build it if he did, and no user has the courage to take out the guides and cut them off or cut away the surface even when he knows it would be money in his pocket, but it is the thing to do. We find that in the case of a slipper guide, owing to the effect of inertia and momentum giving a twisting action to the crosshead, it is necessary to cut away the guide so that the crosshead will over-run very nearly one-half its length before the scraper marks will show uniform wear. This, of course, is subject to modification according as the centre of gravity is higher or lower, or the speed of the engine is greater or less. We are building engines with the crossheads over-running that way and people buy them.

11. To get the best out of machines, they not only want to be rigid and true, but the drive needs to be powerful. In this respect a worm gear is about as perfect as can be, or cutting spur gear teeth spiral accomplishes about the same result. What appears as an objection to spiral teeth is end thrust against the shoulders, which does not amount to much, and when the shaft runs in reverse directions and end play in the journals is permissible, the journals keep in much better condition. The mention of a worm gear is like the flaunting of a red rag to some people, but it has its place and a good many more places than it has been used in. The claimed objection is excessive friction and loss of power, but the results do not seem to justify the claim.

12. The most perfect worm gear we have (theoretically) is a screw and nut, and they do waste enormously in friction, and in proportion to what they do they wear out the most of any piece of mechanism. The most imperfect worm gear we have (theoretically) is the Seller's planing machine drive, and yet they never wear out, and hence cannot lose much in friction.

13. In the writer's opinion two of the things which never need to have been invented are the Hindley worm gear and a machine for hobbing worm gear. Experience convinces the writer that a liberal pitch worm skewed round so as to properly mesh with a

plain spur gear, or one with the teeth at such an angle as to skew the worm a little more will run more easily and last longer than the other sort. A machine driven with the worm is positive, and if there is any chatter it comes from elasticity in the spindle or the work itself. The value of lathes, particularly those used for face plate work, is considerably improved by having large and short main bearings. They should be large to resist torsion and short to resist bending, and the ordinary face plates are ridiculously frail. To get the best of a face plate it should be box section and as large as will swing in the lathe.

14. Owing to the rapid wear of screws the writer is convinced that a precision screw in any lathe used in manufacturing is of no special value over a fairly good one. Wearing the screw in one place while threading a few hundred pieces destroys the precision in a way which no future use will ever correct.

If the designer will analyze every detail he will find that many of the old features were not right to meet the old conditions and not half right for the new.

15. While manufacturing is going to call for many more simple machines—that is, machines to do one thing rapidly and well—the machines which will do a variety of work will be still in demand, for the sparsely settled sections of the country and the colonies will call for the country machine shop as of old.

DISCUSSION.

Mr. John D. Riggs.—There is no question but that there is now a demand for a general re-designing of machine tools, but is this entirely due to the new high-speed steels? Is it not rather due to the fact that the designs were only half right, and now with the new steels this half right is being reduced to one-quarter right? If higher speed is all that is required this can be had at once in most cases by putting a larger pulley on the line shaft or at most by adding a high-speed belt between the line and counter shafts. It may be noted that doubling the speed in this way doubles the available power as well.

Cone pulleys have gotten into bad repute, largely, I think, on account of having too many steps and too small diameters.

Established practice may be given credit for some of the weak points in present machine design. It does not follow that because our grandfathers sawed off the ends of their wooden lathe

beds square that present lathe builders should do the same with cast iron ones. Again those builders who *borrow* designs and devote their best efforts to *getting* money do their part in establishing practice.

In applying the individual electric drive the practice of incorporating the motor into the design of the machine seems to be most commendable. One casting may serve as the frame for both machine and motor and yet the machine may be furnished with or without the motor drive.

In applying motors to radial drills why not place the motor on the radial arm very close to the drill spindle and thus dispense with a considerable portion of the mechanism now used to transmit power from a stationary shaft?

The experience of Professor Sweet with worm gears is certainly quite different from my own. While the Sellers planer drive has proved all right, the ordinary worm gear as used in freight elevators in buildings is not. And the worst feature of these machines is that you seldom know how near right they are or when they will go wrong. The lubrication of the two is essentially different.

Those people who have labored to improve the worm gear for elevators have my sympathy but none of my orders so long as the direct hydraulic elevators are in the market.

Mr. H. P. Fairfield.—Any one that has had much to do with the so-called high-speed steels must have been impressed with the fact that the ordinary 14″ or 16″ engine lathe was lacking in material. About one year ago I put in the hands of some of our students tools forged from one of the prominent high-speed steels, my object being to study the uses of the steel. The students' instructions were to break down the tool if possible, then to reduce speed and feed until a desirable balance was reached. On diameters of about two inches the lathes used were not able to make good, and it was necessary to reduce both the feed and speed to prevent seriously injuring the machine. The trouble seemed to be entirely confined to the head stock and carriage, and after some study of the subject and continued observation, I came to a conclusion that the most of the trouble was a lack of material in the head stock itself. As the diameters turned, a high speed of revolution was needed to bring up the surface speed to the desired point, and to prevent tearing the head stock to pieces a heavy face plate of a size as large as the

lathe could swing, was put upon the nose of the spindle. This seemed to correct much of the trouble and I believe is a desirable thing to do, although it is not usual to use a fly wheel on an engine lathe. So far as observed the bed was not affected in any case.

I would suggest that my conclusions are that the engine lathe needs more material in its head stock, broad surfaces in its carriage, nicely gibbed, a positive drive to its feed works, a less number of steps on the cone and broader belts, and a massive face plate.

Mr. Oberlin Smith.—I think the time has now come when that "anvil principle" that Mr. Porter and I used to talk about some years ago must come to the fore. It is true that high-speed steels simply require more horse-power for the higher speed and not necessarily more torque on the lathe spindle; but as a matter of fact these new steels are probably stronger and will take heavier cuts than other steels, without breaking off the cutting edges. Am I right about that, Mr. Taylor?

Mr. Fred W. Taylor.—No, I think not.

Mr. Smith.—Mr. Taylor says I am not right. I thought he was going to back me up.

Mr. Taylor.—I should like to back you up if I could. But directly the opposite is true. The only advantage which the tool steels containing tungsten or molybdenum in combination with chromium and heated to a high heat according to the Taylor-White process have over ordinary tools is that they will cut from two to four times as fast and therefore do much more work in a given time. The presence of tungsten or molybdenum renders the tools weaker and more brittle in the body of the tool. The cutting edge of the tool is also more brittle than the edge of the old carbon steel tool at usual shop temperatures. The Taylor-White tools, while more brittle at the usual temperatures of the air from 50° to 100° Fahr., have the peculiar property of remaining about as hard as they ever were when heated by the friction of the chip which they are cutting up to the extraordinary temperature of 1000° to 1200°; while tools not so treated soften and crumble away when heated to 400° upwards.

Mr. Smith.—When I get through I want Mr. Taylor to tell you what I was trying to tell you, namely, that we need more strength to our machines as well as more speed. One reason is perhaps that the higher speeds cause more vibration in the thin,

fiddle-like castings generally used which are attuned only too well to the new rapidity of motion. At any rate, we all know that we want very much stronger tools. It may be remembered that some years ago I told this Society that the proper way to design a lathe bed was to build it all up in a solid chunk and then modify it a little by putting a slit through the middle to let the chips fall through. I believe I said also that if lathes were from three to four times heavier than they are now there would be a great deal more work turned out in them. If we would make our lathe heads, too, in great masses of solid material it would be all the better. Nobody has yet been brave enough to make a really heavy lathe, but in my opinion there is going to be a tremendous revolution in this respect in the next ten years, in all of our machine-tools. None have had the courage to go at it yet, but it is going to come—but gradually, like all great developments. Another thing we are going to do is to use milling machines very much more than we do now in the place of planers. This is because of the great defect in all planers of moving the heavy weight of the table plus the work. Thus we must fight inertia in both stopping and starting. Another defect of planers as now built is cutting in one direction only. All this is going to bring us to contrive new forms of milling machines. Planers will remain, of course, but they will be modified, in many cases by making the tool move rather than the work, thus following the general principle used in shapers more than we do at present. How all this will develop we cannot see just now, but it is bound to come. We cannot use a high-speed tool at high speed on an ordinary planer, and for that reason, and for other reasons that I might mention, the present machine is likely to become somewhat obsolete in the near future.

*Prof. John E. Sweet.**—I am sorry that more time was not available for discussion. As to the worm gear's not proving satisfactory for elevators I was not conscious of the fact. If I were in the business I would not abandon them until I had skewed around a worm in a spur gear and tried *that.*

*** Author's closure under the rules.**

No. 1015.*

AIR MOTORS AND AIR HAMMERS.

APPARATUS AND METHODS FOR TESTING.

BY MAX H. WICKHORST, AURORA, ILL.
(Junior Member of the Society.)

1. The apparatus and methods described below are those used in some extensive tests of air-drill motors and air hammers made by the Chicago, Burlington and Quincy Railroad Co. in its laboratory at Aurora, Ill. The tests were made for the purpose of determining the air consumption, horse-power, stalling load of motors, number and force of hammer blows, etc.

General Arrangement.

2. The general arrangement of the apparatus is shown in Fig. 18 and in photograph, Fig. 19.

The air used for making the tests was obtained from the shop supply, which was generally about 60 or 70 pounds. As we desired pressures varying from 60 to 120 pounds, we stepped up the pressures by pumping the air through a 9-inch Westinghouse pump which had the air cylinder bushed to a diameter of 7 inches. This bushing was necessary as the steam pressure at the Laboratory is only about 60 pounds. The air was then pumped into reservoir No. 1, with ordinarily a pressure of 140 or 150 pounds. From here the air was allowed to flow into reservoir No. 2, where it was maintained at any pressure desired by means of a reducing valve, which was a regular 1-inch Westinghouse air-pump governor. An oil cup containing a thermometer was screwed into reservoir No. 2, and was used to determine the temperature of the air supplied to the tool. Another thermometer was also used to note room temperature.

---

* Presented at the New York meeting, December, 1903, of the American Society of Mechanical Engineers, and forming part of Volume XXV. of the *Transactions.*

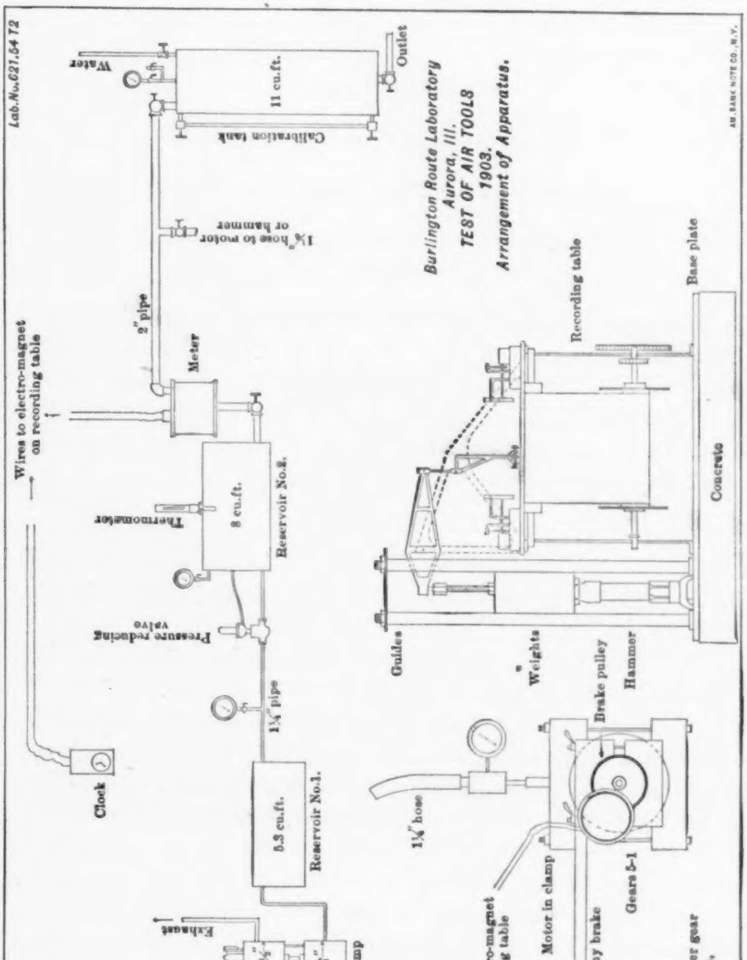

Lab. No. 621.54 T2
Wires to electro-magnet on recording table
Clock
Steam
Exhaust
9½"
7"
Pump
¾" pipe
From air main
5.3 cu.ft.
Reservoir No.1.
1¼" pipe
Pressure reducing valve
8 cu.ft.
Reservoir No.2.
Thermometer
Meter
2" pipe
1¼" hose to motor or hammer
Water
11 cu.ft.
Calibration tank
Outlet
Burlington Route Laboratory
Aurora, Ill.
TEST OF AIR TOOLS
1903.
Arrangement of Apparatus.
Recording table
Base plate
Concrete
Guides
Weights
Hammer
Brake pulley
1¼" hose
Motor in clamp
Gears 5-1
Wires to electro-magnet on recording table
Prony brake
Platform of scales
Contact on larger gear
Wiper on brake.
Wickhorst, M.E.
AM. BANK NOTE CO., N.Y.

FIG. 18.

3. From the reservoir No. 2 the air passed through a meter. This is a high-pressure meter made by the Equitable Meter Co., Pittsburg, Pa., and in construction is similar to an ordinary gas meter, the air alternately filling out and exhausting from leather bellows. From here the air was delivered to the tool to be tested, through a 2-inch pipe and $1\frac{1}{4}$-inch hose. At the point where the air was delivered to the tool we had an expansion consisting of a

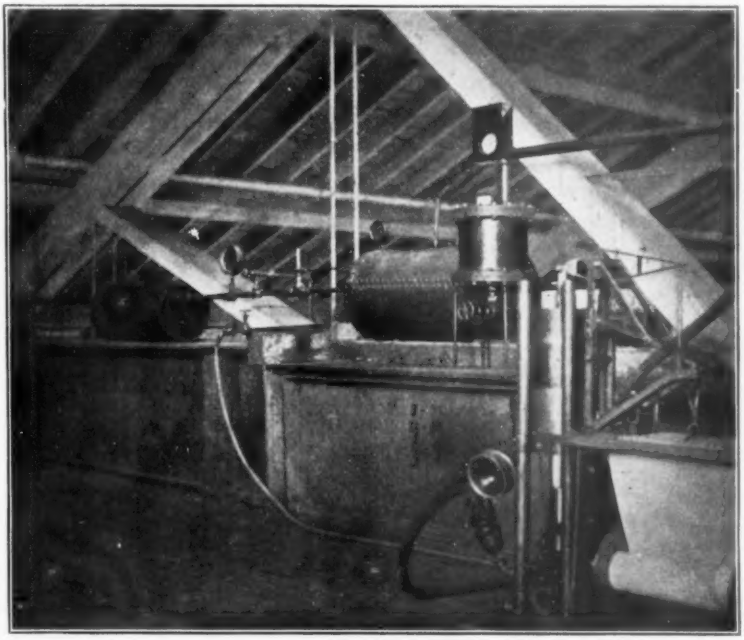


FIG. 19.

2-inch Tee, with an internal diameter of about $2\frac{1}{4}$ inches, in which we determined the pressure by means of a gauge. Care was taken that between the point where the pressure readings were taken and the tool to be tested, there was no contraction in the supply pipe smaller than the opening into the tool under test.

4. For the purpose of calibrating the air meter we used the tank as shown in Fig. 18. This tank had a gauge glass its full length, and its cubic capacity was determined for each 5 inches on the glass by weighing the water.

5. The records of time, revolutions of motor, air consumed and strokes of hammer were obtained autographically, using the record

table shown in Fig. 18. The records were obtained on glazed manilla paper 14 inches wide, moving across the table under electro-magnetic pens. The driving mechanism of the paper consisted of an air motor and suitable gearing. The electro-magnets actuated stylographic pens feeding red ink. One of the pens was actuated by a clock, making a contact every five seconds. Another was actuated by the air meter, a wiper making contact with the teeth of one of the gear wheels in the recording mechanism and each contact representing about $\frac{1}{4}$ cubic foot of air. The third pen was used to record the revolutions of the motors by arranging a wiper and a simple gearing, so as to make a contact every 5 revolutions of the socket for holding the drill.

6. In calibrating the meter the method was to have the calibration tank about full of water, the valve from the meter opened up, thus allowing full pressure; the outlet valve was opened, the record paper started going and as the water in the gauge glass passed the marks 5 inches apart record was made by the observer pressing a push button. The meter at the same time made its own record, and thus we were able to figure out the number of cubic feet per contact or per notch. We also obtained record of 5-second intervals. As the readings of the meter varied somewhat with different rates of flow, calibrations were made at different rates by varying the opening at the outlet valve. A calibration curve was then made by plotting notches per minute as abscissæ and cubic feet per notch as ordinates. A number of calbration tests were made during the course of the tests of the tools.

The various gauges used from which pressure readings were taken were previously checked up and adjusted by means of a Crosby Dead Weight Tester.

Motor Tests and Calculations.

7. The arrangement used for testing motors is also shown in Fig. 18 and in the photograph, Fig. 20. The arrangement was to apply the load by means of a Prony Friction brake, the revolutions being recorded by means of a wiper making contact every five revolutions. The air consumption and time were recorded as described above.

8. The procedure was to first regulate the pressure in reservoir No. 2 at 60 pounds, then put on a light brake load, keeping this constant during the test with full open throttle. Another test

was then made with heavier brake load with the same air pressure, and the increments of load continued in successive tests till the tool was stalled. Then air pressures of 80, 100 and 120 pounds were used in the same manner. A sample of a motor record reduced is shown in Fig. 21.

9. Table No. 1 shows a sample data sheet, and the various items and calculations were obtained as follows:

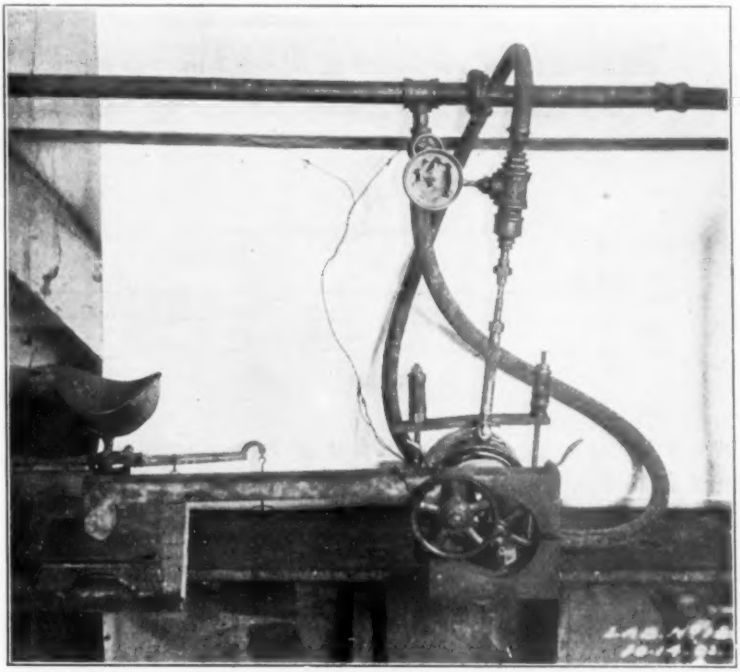


FIG 20.

No. 1 and No. 2 are pressures as read by an observer on gauges and are pressures above atmospheric pressure.

No. 3 is temperature in degrees Fahrenheit of the compressed air in reservoir No. 2.

No. 4 is the meter contacts or notches obtained from the autographic record included in a strip of record covering one minute as recorded by the clock.

No. 5 is the cubic feet of compressed air per meter notch as obtained from the calibration chart.

No. 6 is the cubic feet of compressed air per minute obtained by multiplying Nos. 4 and 5.

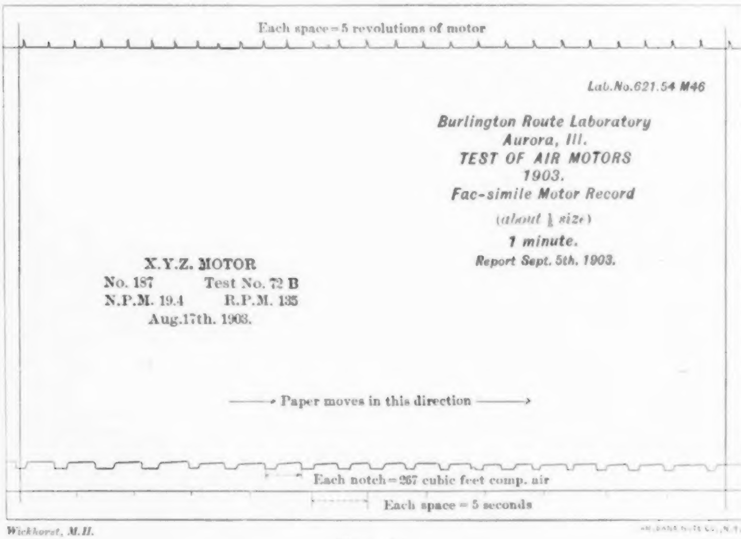


FIG. 21.

No. 7 is the number of cubic feet of free air per minute obtained from No. 6 by the following formula:

$$F\,A = \frac{C\,A \times (P + 15)}{15}$$

where $F\,A$ = free air in cubic feet per minute.
$C\,A$ = compressed air in cubic feet per minute.
P = pounds gauge pressure at tool.

No. 8 is the revolutions per minute of socket for holding the drill and is obtained from the autographic record.

No. 9 is the brake load as shown by the weight on the scale at the end of the lever arm, usually three feet, except with the smaller motors, where the lever arm was two feet. This brake load was pre-determined and kept constant by an observer during each test.

No. 10 is the brake horse-power, and was calculated as per following formula:

$$B\ H\ P = \frac{R\ P\ M \times 2r \times 3.1416 \times w}{33000}$$

where $B\ H\ P$ = brake horse-power.
$R\ P\ M$ = revolutions per minute of brake-wheel.
r = radius in feet of brake-lever.
w = weight in pounds on scale.

No. 11 is the load on scale which was just sufficient to stall the tool.

No. 12 is the stalling load at one foot radius calculated from No. 11.

No. 13 is the cubic feet of free air consumed per minute per horse-power, obtained by dividing No. 7 by No. 10.

10. After obtaining these various data we plotted three curve

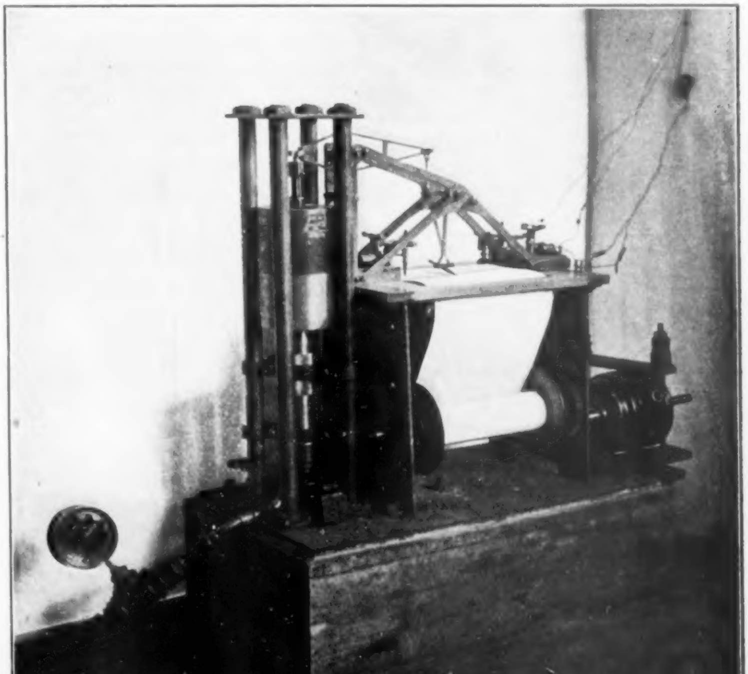


FIG. 22.

TABLE I.

BURLINGTON ROUTE LABORATORY. Lab. No. 621.54 M 47.
AURORA, ILL.

TEST OF AIR MOTORS,
1903.
Data Sheet.

AIR MOTOR TEST. Test 61.

MAKER: Air Tool Co. NAME OF TOOL: Mendota. SIZE: 3. TYPE: 3 Cyl.

WEIGHT: 47 Lbs. SERIAL NUMBER: 82. COST:

REMARKS:

BRAKE LEVER ARM: 3 ft.

DATE: 8/4-03. TEMPERATURE ATMOSPHERE: 87 deg.

	A	B	C	D	E	F	G	H	I
1. Press. reservoir	101	101	101	101	121	121	121	121	121
2. " at tool	100	100	100	100	120	120	120	120	120
3. Tempr. comp. air	101	102	102	103	106	108	110	112	114
4. Notches per min.	24.	21.2	19.5	17.6	23.4	24.	20.	20.3	17.5
5. Cu. ft. per notch	.262	.264	.265	.266	.263	.262	.264	.264	.266
6. Cu. ft. C. A. min.	6.8	5.6	5.2	4.7	6.2	6.8	5.3	5.4	4.6
7. Free air	52.	42.8	40.	36.	56.	61.3	48.	47.7	41.5
8. Rev. per min.	138	119	89	72	135	115	91	88	78
9. Wt. on scale	20	25	30	35	25	30	35	40	45
10. Brake H.-P.	1.57	1.7	1.52	1.45	1.92	1.97	1.82	2.02	2.
11. Stalling weight	.....	.....	.....	37	.....	.....	.....	.....	48
12. "11" at one ft. radius	.....	.....	.....	111	.....	.....	.....	.....	144
13. "Seven" per H.-P.	33	25	26	24.8	29	31	26.5	23.5	20.7

(Signed) ....

Observer.

sheets to show up the results by the graphic method, representing in each case the air pressure as abscissæ. On one we plotted as ordinates the stalling load at one foot radius, on another the air consumption in cubic feet of free air per minute per horse-power at maximum horse-power, and on the other the maximum horse-power.

Air Hammer Tests and Calculations.

11. The arrangements for testing air hammers is also shown in Fig. 18 and photograph, Fig. 22. The method in general was to let

FIG. 23.

the hammer strike upward against a known weight adjusted to the size of the hammer and to autographically record the distance the weight was lifted. The weights varied from about 40 to 150 pounds, and the vertical lift was multiplied 8 times on the record. The time and air consumption were recorded as above described, and a sample of one of the records obtained is shown in Fig. 23.

12. The results of test were recorded on a blank, copy of which is shown in Table II.

No. 1 and No. 2 are gauge-pressure readings as noted by an observer.

TABLE II.

Lab. No. 621.54 H 12
Test 501.

AIR HAMMER TEST.

MAKER : Air Tool Co. NAME OF TOOL : Vulcan. SIZE : 8.

WEIGHT : 22 lbs. SERIAL NUMBER : 3171. COST :

WT. PLUNGER : 1 lb. DIAM. CYLINDER : $1\frac{1}{16}''$. STROKE INCHES : 8.

REMARKS :

DATE : 7/22-03. TEMPERATURE ATMOSPHERE : 87 deg.

	A	B	C	D	E	F	G	H	I
1. Press. reservoir	60	80	100	120					
2. " at tool	60	80	100	120					
3. Temp. comp. air	88	90	90	91					
4. Notches per min	29	24	25	21					
5. Cu. ft. per notch	.271	.269	.27	.268					
6. Cu. ft. C. A. min	7.85	6.45	6.75	5.63					
7. Free air	39.2	40.8	51.6	50.6					
8. Strokes per min	834	892	964	1,000					
9. Weight	120	120	120	120					
10. Distance raised	.004′	.0052′	.006′	.0065′					
11. Ft. lbs. per blow	.48	.624	.72	.7818					
12. Horse-power	.0122	.0168	.021	.0236					
13. "Seven" per H.-P	3,200	2,400	2,400	2,100					

(Signed) ....
Observer.

BURLINGTON ROUTE LABORATORY
AURORA, ILL.
TEST OF AIR HAMMERS,
1903.
Data Sheet.

No. 3 is the temperature in degrees Fahrenheit of the compressed air in reservoir No. 2 as noted by an observer.

No. 4 is the meter contacts or notches per minute as obtained from the autographic record.

No. 5 is the cubic feet of compressed air per notch as obtained from the calibration chart.

No. 6 is the cubic feet of compressed air per minute obtained by multiplying Nos. 4 and 5.

No. 7 is the cubic feet of free air per minute obtained by the same formula as given under air motors.

No. 8 is the number of strokes per minute obtained from the record by counting the strokes made in one minute.

No. 9 is the pounds of the weight placed over the hammer.

No. 10 is the average distance in feet the weight was raised as obtained from the record.

No. 11 is the foot pounds of effective work per blow, obtained by multiplying the weight in pounds by the distance in feet it was raised.

No. 12 is the horse-power of the hammer, obtained as per following formula:

$$HP = \frac{\text{ft. lbs.} \times \text{blows per min.}}{33000}$$

No. 13 is the cubic feet of free air per minute per horse-power.

13. After obtaining the results of test, three curves were plotted from them as follows, in each case the pounds gauge pressure being plotted as abscissæ and the following as ordinates: foot pounds per blow, cubic feet of free air per minute per horse-power and horse-power.

14. In conclusion, the author desires to express his special thanks and acknowledgments to Mr. H. F. Wardwell, who did the greater part of the work in making the designs and tests.

DISCUSSION.

Mr. Frank G. Hobart.—The method here described of arranging the hammer to strike against a weight and basing calculations upon the displacement of this weight would seem useful only for obtaining comparisons between different hammers.

The facsimile diagram shows a movement of the pencil of about

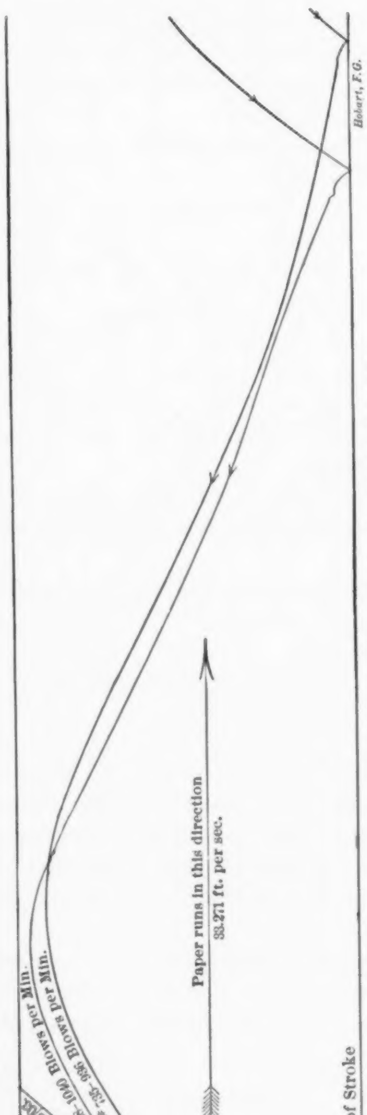


DIAGRAM OF #6 RIVETING HAMMERS #735 & #738. 100 LBS. AIR PRESSURE.

Fig. 24.

$\frac{1}{8}$ inch, corresponding to a movement of the weight of not more than $\frac{1}{64}$ inch. With this small movement, together with the long lever reductions between the weight and the pencil, the high speed and the various questions about impact, weight, friction, etc., I should not expect the figures based on the diagram could be accurate.

Fig. 24 shows two diagrams made on one sheet of paper by $1\frac{1}{16}$ x 6 inch riveting hammers on a machine which we are using for air-hammer testing. It shows the number of blows per minute, length of stroke of the pistons, velocity of pistons at every portion of the stroke and the position of the pistons at every instant. Knowing the weight of the piston, its velocity at impact and the number of blows per minute, the ability of the hammer to do work can be calculated. The effect of changes of port areas or other details of the hammers is very clearly shown by such diagrams.

The machine consists of a drum driven by a small air motor at a periphery speed of about 2,000 feet per minute. A centrifugal governor holds this speed very nearly uniform, and a paper wound on this drum receives the diagram. The hammer is mounted with its length parallel to the axis of the drum, and in such position that a slender steel rod connects the piston to a pencil slide which marks the position of the piston on the diagram. All of the moving parts are very delicate and do not interfere, so far as can be detected, with the action of the piston. The machine was built for experimental work and for testing hammers by Fairbanks, Morse & Co. for use at their works at Beloit, Wis.

*Mr. M. H. Wickhorst.**—Commenting on Mr. Hobart's remarks, I would say the tests were made for the purpose of comparing the various hammers on the market, and I wish to correct his impression concerning the distance the weight was lifted in the tests. The sample record as printed is only a little over one-fifth size. The method of studying a hammer which he describes, I should think, would be decidedly valuable to a designer, and am very glad that he has presented to the Society an outline of the apparatus and the sample record.

---

* Author's Closure under the Rules.

No. 1016.*

A METHOD FOR DETERMINING RATES AND PRICES FOR ELECTRIC POWER.

BY FRANK B. PERRY, BOSTON, MASS.
(Junior Member of the Society.)

1. That the future will show a rapid growth in the application of electricity to industrial pursuits, operated from a centrally located steam-driven electric power station, has led me to present the following data to our members, many of whom will doubtless find the methods suggested applicable to their own interests. The advent of driving textile and other mills from a plant of this character has opened a question for discussion as to the establishment of a proper basis for rates to be charged for electric current supplied in large quantities.

2. The contracts drawn up by electric companies with their customers sometimes contain a clause similar to the following, which is given for the purpose of illustration, and is not intended to cover any specific case: For a period of years from date the lessor agrees to furnish, and the lessee to receive and to pay for, within the times and on the terms set forth, all the power that may be required to properly operate and light his plant. The amount of power to be determined by meter readings, and to be billed monthly at the rates recited below, viz.:

1,800 to not more than 2,160 kilowatts at rate of $31.50 per kilowatt per annum;

More than 2,160 kilowatts and not exceeding 2,520 kilowatts at rate of $30.00 per kilowatt per annum;

More than 2,520 kilowatts and not exceeding 2,700 kilowatts at rate of $28.50 per kilowatt per annum;

---

* Presented at the New York meeting (December, 1903) of the American Society of Mechanical Engineers, and forming part of Volume XXV. of the *Transactions*.

All in excess of 2,700 kilowatts at rate of $27.60 per kilowatt per annum.

3. The above rates are based on an annum of 3,000 hours. From this schedule the following tables are derived:—

TABLE I.

Rate per Kw. per Annum.	Equiv. Rate per H. P. per Annum.	Rate per Kw. per Hour.	Rate per H. P. per Hour.
$31.50	$23.499	$ .0105	$ .007833
30.00	22.38	.0100	.007460
28.50	21.26	.0095	.007087
27.60	20.5896	.0092	.006832

TABLE II.

Kilowatts Used.	Cost per Annum.	Cost per Month.
1,800	$56,700.00	$4,725.00
2,160	68,040.00	5,670.00
2,161	64,830.00	5,402.50
2,268	68,040.00	5,670.00
2,520	75,600.00	6,300.00
2,521	71,848.50	5,987.37
2,652.7	75,601.95	6,300.16
2,700	76,950.00	6,412.50
2,701	74,547.60	6,212.30

Table I. is self-explanatory. An examination of Table II. shows plainly the faults that exist in a schedule such as that usually followed.

4. By using 2,161 kilowatts, or one additional kilowatt more than 2,160 kilowatts, a yearly saving of $3,210 may be made; similarly an increase of one kilowatt above 2,520 kilowatts effects a reduction of $3,751.50 per annum, and one extra kilowatt over 2,700 kilowatts lessens the yearly expense by $2,402.40.

5. While these figures show the points in the schedule at which the greatest saving may be made, many other quantities which are given in a later table illustrate equally well the irregularity of prices based on such a list of rates. The inconsistency of this method is further shown by the fact that 2,268 kilowatts cost the same as 2,160 kilowatts, or 108 kilowatts may be utilized without any increase in expense to the consumer. The price is also practically the same for 2,652.7 kilowatts as for 2,520 kilowatts, which indicates that any number of kilowatts between these limits may be used without additional cost to the person buying power.

6. It would appear from these figures that the electric company makes more profit at some places in the schedule than at others,

and on that account could furnish certain amounts of power gratis. Of course, this is not true, but tends to convince one of the absurdity of the system that is customarily used by central electric stations. It is also evident that the customer, if he were so disposed, might watch his meter and so adjust his consumption of current as to bring about a substantial decrease in his bill for the month. For example, he might install a few extra machines that would serve not only to increase his product but

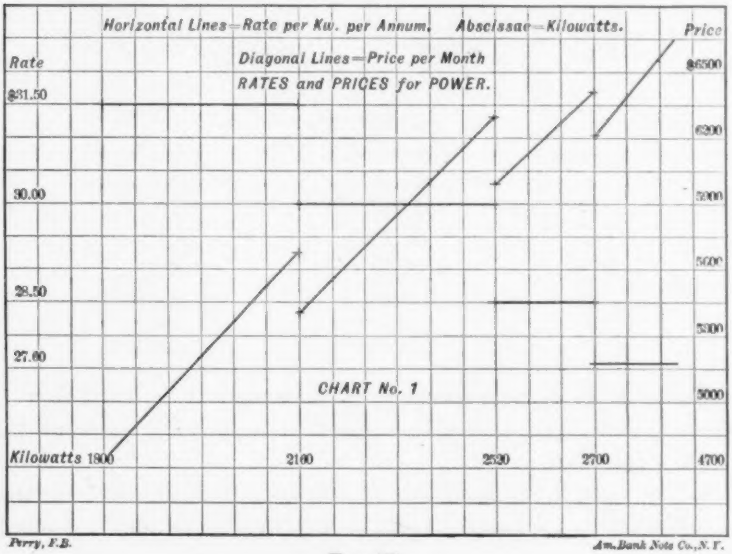


FIG. 25.

would also be the means of diminishing his bill. As an extreme case, the use of one additional kilowatt would bring about this result. It also might prove economical to burn electric lights during the day in order to increase the current consumption.

7. Both of these examples mentioned serve as illustrations of the fact that it is within the consumers jurisdiction, by the terms of his agreement, to use power advantageously with a resultant financial loss to the electric company. While such a procedure may be perfectly legitimate, it would be fairer to all concerned to prevent contingencies of this kind from arising, and at the same time to provide a system of rates which would be to the mutual interests of both parties.

8. The horizontal lines on Chart No. 1, Fig. 25, indicate the rates

for power, and the diagonal lines represent the price per month for electric current covering the entire range of the above mentioned schedule. The latter lines show at a glance the unfairness of the system, and prove conclusively that rates cannot be adjusted properly on the customary basis.

9. The subsequent discussion is intended to give a method for rearranging a schedule of rates based on the "step-system," and for convenience the aforesaid figures will be used. The principles, however, may be applied to any other rates made up in a similar manner.

10. Referring to Table II. and applying the well-known formula $(a + l)\frac{n}{2}$, giving the sum of the terms of an arithmetical progression we obtain for each step, assuming a common difference of one kilowatt, the following figures:

	Value of "a."	Value of "l."	Value of "n."
(1) 2,161 – 2,520 kw........	$64,830.00	$75,600	360
(2) 2,521 – 2,700 "	71,848.50	76,950	180

11. The value of "*a*," or the first term of the series represents the product of 2,161 kilowatts by its rate per kilowatt per annum. In a similar way the other terms of the series are obtained, "*l*," indicating in each case the last term and "*n*" the number of terms.

12. Since the rate is constant from 1,800 to 2,160 kilowatts, we will compute simply the sum of the series beginning at 2,160 kilowatts.

Sum of (1) = (64,830 + 75,600) 180 =	$25,277,400
" " (2) = (71,848.5 + 76,950) 90 =	13,391,865
2,160 kw. at $31.50 =	68,040
(3) Total sum of two series =	$38,737,305

13. With a varying rate of power, diminishing in a fixed ratio between any two amounts, it is possible, by the substitution of proper quantities in an equation herein deduced by the writer, to compute the value of the sums of the product of each succeeding kilowatt by its corresponding rate.

14. The horizontal spaces of the charts, or abscissæ, represent in terms of kilowatts, a progressive increase in each successive number by the addition of any desired equal amount.

Let (a) = 1st term. (l) = last term. (d) = common difference (n) = number of terms. (s) = sum of terms.

Then 1st term $= a$,
2d " $= a + d$,
3d " $= a + 2d$,
4th " $= a + 3d$,

$$l = a + (n - 1)d,$$

$$s = (a + l)\frac{n}{2}.$$

15. The vertical spaces of charts, or ordinates, represent in terms of rates per kilowatt annum, or per kilowatt hour, a progressive decrease in each successive number by the subtraction of any chosen equal quantity.

Let (b) = 1st term. (k) = last term. (r) = common difference. (t) = sum of terms. (m) = number of terms.

Then 1st term $= b$,
2d " $= b - r$,
3d " $= b - 2r$,
4th " $= b - 3r$,

$$k = b - (m - 1)r,$$

$$t = (b + k)\frac{m}{2}.$$

16. Multiplying the corresponding terms of these two series, we obtain the following:—

(4) Product of 1st term $= ab$,
(5) " " 2d " $= (a + d)(b - r)$,
(6) " " 3d " $= (a + 2d)(b - 2r)$,
(7) " " 4th " $= (a + 3d)(b - 3r)$,
(8) " " last " $= \{a + (n - 1)d\}\{b - (m - 1)r\}$.

17. Adding these products, and developing the quantities enclosed in parenthesis, we obtain for the sum of 1st and 2d terms designated (4) and (5).

(*A*) Sum of products 1st and 2d terms $= 2ab + bd - ar - dr$.

Multiplying equation by $\frac{2}{2}$ $= \frac{4ab + 2bd - 2ar - 2dr}{2}$.

Substituting values $n = 2$ and $m = 2$ in the above, we find

$$(A) = \frac{2abn + bdn - arm - drm}{2}$$

$$= \frac{bn}{2}\left\{2a + (n - 1)d\right\} - \frac{rm}{2}(a + d).$$

18. In a similar manner $(B) =$ sum of (4), (5) and (6),
and $(C) =$ " " (4), (5), (6) and (7),

$$(B) = \frac{bn}{2}\left\{2a + (n-1)d\right\} - \frac{rm}{2}(2a + 3\tfrac{1}{3}d),$$

$$(C) = \frac{bn}{2}\left\{2a + (n-1)d\right\} - \frac{rm}{2}(3a + 7d).$$

19. Comparing equations (A), (B) and (C), we find that they are alike in all respects excepting the coefficients of "a" and "d" of the negative terms.

20. Substituting these values, viz., for coefficient of "a" $(m - 1)$, and for coefficient of "d" $\left\{\frac{2}{3}(m-2)+1\right\}\left\{m-1\right\}$, we have for the sum of any number of products containing "n" and "m" terms

$$\text{Total sum} = \frac{bn}{2}\left\{2a + (n-1)d\right\}$$
$$- \frac{rm}{2}\left\{a + \left[\frac{2}{3}(m-2)+1\right]d\right\}\left\{m-1\right\}.$$

$$(D) \quad \text{"} \quad \text{"} \quad = \frac{bn}{2}(a + l) - \frac{rm(m-1)}{2}\left\{a + \frac{d}{3}(2m-1)\right\}.$$

21. In order to apply this formula to the case in question, let us consider the rate as beginning at \$31.50 per kilowatt per annum for 2,160 kilowatts and varying in amount to \$27.60 per kilowatt per annum for 2,700 kilowatts. By so doing, we preserve the limiting features of the schedule both for rates and for the quantities at which they are applicable. Since the successive steps in the first mentioned schedule show an even decrease in rate for equal increments of power up to 2,520 kilowatts, it is safe to assume that a list of rates may be made up to vary uniformly from 2,160 to 2,520 kilowatts and regularly, but at a different rate, from 2,520 to 2,700 kilowatts.

22. Let $x =$ rate per kilowatt per annum for 2,520 kilowatts.

$a_1 = 2{,}160$	$a_2 = 2{,}521$
$b_1 = 31.50$	$b_2 = x$
$d_1 = 1$	$d_2 = 1$
$l_1 = 2{,}520$	$l_2 = 2{,}700$
$r_1 = \dfrac{(31.5 - x)}{360}$	$r_2 = \dfrac{(x - 27.6)}{180}$
$n_1 = 361$	$n_2 = 180$
$m_1 = 361$	$m_2 = 180$

23. Substituting these respective values in equation (D) gives

$$D_1 = \frac{31.5 \times 361}{2}(2{,}160 + 2{,}520) - \frac{(31.5 - x) \times 361 \times 360)}{2 \times 360} \{2{,}160 + \tfrac{1}{3}(722 - 1)\}.$$

$$D_2 = \frac{180x}{2}(2{,}521 + 2{,}700) - \frac{(x - 27.6) \times 180 \times 179}{2 \times 180} \{2{,}521 + \tfrac{1}{3}(360 - 1)\}$$

From the above,

$$D_1 = \frac{159{,}655{,}860 - 81{,}886{,}171.5 + 2{,}599{,}561x}{6}.$$

$$D_2 = \frac{2{,}819{,}340x - 1{,}418{,}038x + 39{,}137{,}848.8}{6}.$$

24. Then in order to fulfill the conditions of the first named schedule, $D_1 + D_2$ must equal (3) or

$$D_1 + D_2 = 38{,}737{,}305.$$

Substituting values of D_1 and D_2 and solving,

$$4{,}000{,}863x = 115{,}516{,}292.7$$
$$x = 28.872.$$

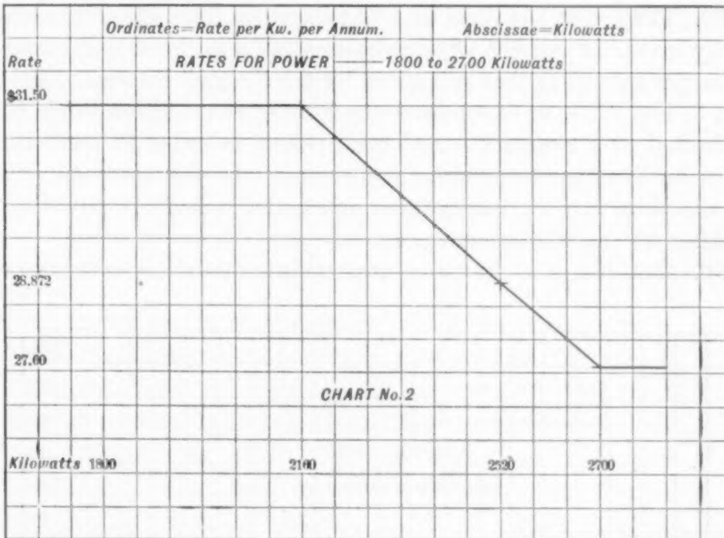
25. This value of "x" substituted in expressions for rates gives $r_1 = \$.0073$ and $r_2 = \$.00706\frac{2}{3}$ as the variations in charges for each kilowatt per annum from 2,160 to 2,520 kilowatts, and from 2,520 to 2,700 kilowatts respectively.

26. The rates per kilowatt hour become $r_1 = \$.0000024\frac{1}{3}$, $r_2 = \$.0000023\frac{5}{9}$; for 30 kilowatts hours, $r_1 = \$.000073$ and $r_2 = \$.0000706\frac{2}{3}$.

27. Using this system as a basis for rates, the wording of the agreement for power would be changed to read as follows:

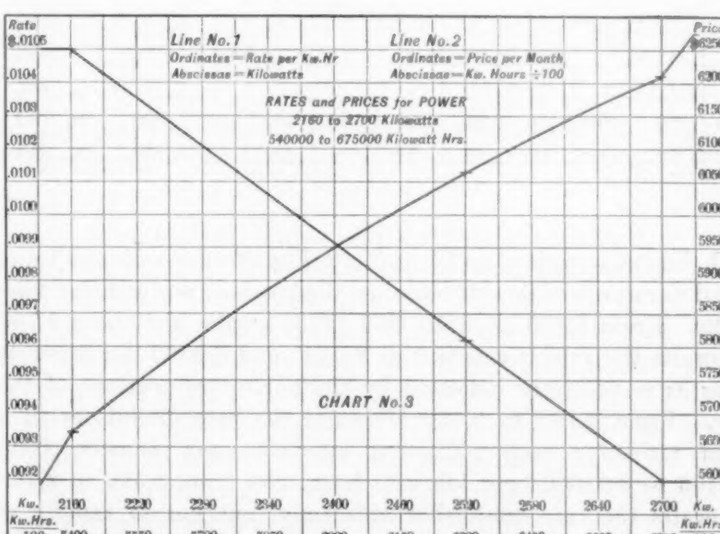
"The charge for electric current furnished under this contract shall be made as per the chart attached hereto, and made a part hereof."

28. The line plotted on Chart No. 2, Fig. 26, indicates the rates per kilowatt per annum which would, in this instance, replace the original schedule as given by Chart No. 1, Fig. 25.



Perry, F.B. Am. Bank Note Co., N.Y.

FIG. 26.



Perry, F.B. Am. Bank Note Co., N.Y.

FIG. 27.

29. The results obtained by following these lines of rates necessarily gives, on account of the above reasoning, the same average price per month as that computed by the "step" system. Chart No. 3, Fig. 27, is an example of the form of chart which would be embodied in a contract for electric power in order to meet the rates outlined in the original agreement. It also gives an idea of the simplicity of interpreting rates and prices for power by making use of the scheme proposed.

30. Line No. 1 is drawn through points plotted in accordance

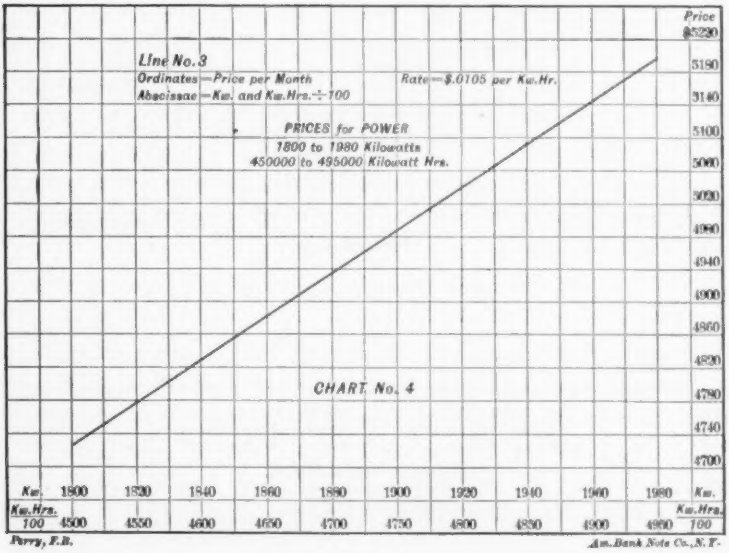


FIG. 28.

with the rates specified on Chart No. 2, Fig. 26; the ordinates being equal to rates per kilowatt hour and the abscissæ representing kilowatts. From 1,800 kilowatts to 2,160 kilowatts and above 2,700 kilowatts the rate is constant at $.0105 and $.0092 per kilowatt hour as respectively indicated by the horizontal portions of the line. From 2,160 to 2,520 kilowatts the rate diminishes in a fixed ratio for each additional kilowatt used, namely, from $.0105 to $.010062 per kilowatt hour; also from 2,520 to 2,700 kilowatts the rate diminishes uniformly, but not so rapidly as between the points previously stated, from $.010062 to $.0092 per kilowatt hour. It will be noted that the ordinates may be

read easily to $\frac{1}{1000}$ of a cent per kilowatt hour and the abscissæ to 3 kilowatts without estimating fractional division of the spaces.

31.* Line No. 2 is deduced from the foregoing, and by its means one may determine at a glance, knowing the wattmeter reading, the amount of each month's bill for power consumed.

32. The lower line of figures on the chart represent meter readings in kilowatt hours divided by one hundred. These quantities are computed by multiplying the kilowatt readings by 250, the average hours per month, or $\frac{1}{12}$ the total for the year as limited by

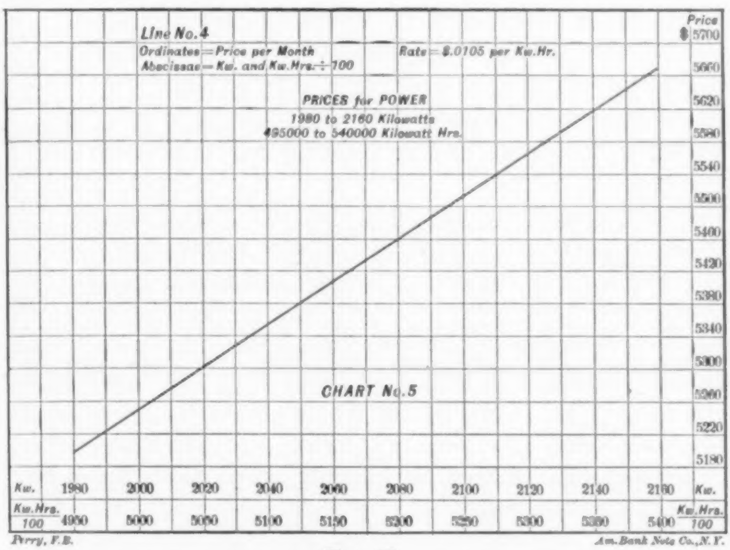


FIG. 29.

the contract. Two ciphers are omitted from these results to avoid repetition of figures. To obtain the bill per month, find the number on chart corresponding to that recorded by the wattmeter, and note the point vertically over same on line No. 2. In the column at the right-hand side of the chart, horizontally opposite this intersection, will be found the price of bill for the amount of power in question. The abscissæ may be read accurately to 750 kilowatt hours and the ordinates to $5 for the corresponding spaces, and by estimation of tenths these differences may be proportionately reduced.

33. Charts No. 4 to No. 8 (Figs. 28 to 32) inclusive are similar to, and may be considered as supplementary to chart No. 3. Each

chart includes a section of 180 kilowatts or $\frac{1}{5}$ of the entire range from 1,800 to 2,700 kilowatts.

34. Lines No. 3 and No. 4 indicate prices for power up to and including 2,160 kilowatts at a constant rate of \$31.50 per kilo-

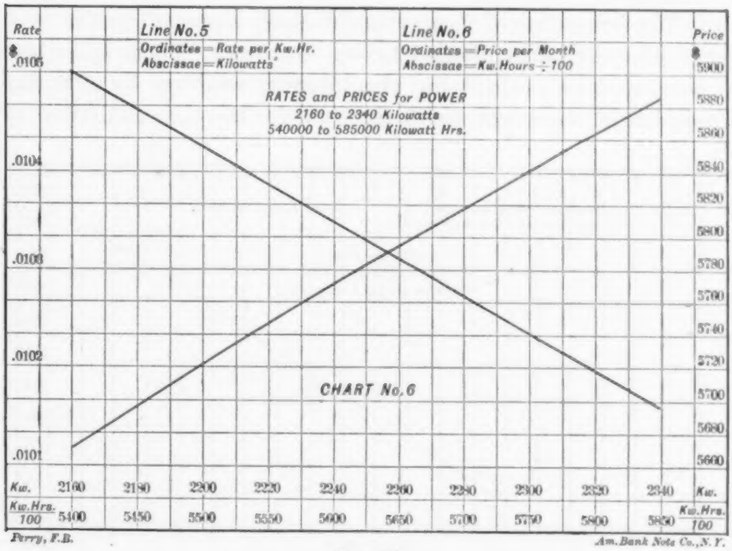


FIG. 30.

watt per annum, or \$.0105 per kilowatt hour. Lines No. 5, No. 7 and No. 9 represent rates

varying respectively from	\$31.50	to	\$30.186	per	kw.	per	annum.
" " "	30.186	"	28.872	"	"	"	"
" " "	28.872	"	27.60	"	"	"	"
or from	\$.0105	to	\$.010062	per	"	"	hour.
	.010062	"	.009624	"	"	"	"
	.009624	"	.0092	"	"	"	"

35. Lines No. 6, No. 8 and No. 10 show prices per month for power, varying in the order given from 2,160 to 2,340 kilowatts, from 2,340 to 2,520 kilowatts and from 2,520 to 2,700 kilowatts. Each of the spaces on the charts No. 4 to No. 8 inclusive occupied by the abscissæ represent 1 kilowatt or 250 kilowatt hours as the case may be, consequently are well within the limit that it is possible to read a recording wattmeter of 3,000 kilowatts capacity.

36. Greater accuracy is also attainable in determination of rates per kilowatt hour or the total bill for the month. This may be carried to a still further degree of refinement by an increased subdivision of the amounts of power and their accompanying rates. The principles enumerated may be applied equally well to a given list of powers with any predetermined rates. The Table III. is presented to show the comparison of prices per month that would be paid for various amounts of power by the original agreement, and also by the lines illustrated on charts No. 3 to No. 8 inclusive (Fig. 27 to 32).

TABLE III.

Equivalent H. P.	Kws.	Kw. Hrs. per month of 250 Hrs.	Rate per Kw. Hr.	Monthly Bill.		Difference.
				By Chart.	By Schedule.	
2,894.1	2,160	540,000	$.0105	$5,670.00	$5,670.00	— ....
2,935.7	2,190	547,500	.010427	5,708.78	5,475.00	— $233.78
2,977.2	2,220	555,000	.010354	5,746.47	5,550.00	— 196.47
3,016.1	2,250	562,500	.010281	5,783.06	5,625.00	— 158.06
3,056.3	2,280	570,000	.010208	5,818.56	5,700.00	— 118.56
3,096.5	2,310	577,500	.010135	5,852.96	5,775.00	— 77.96
3,136.7	2,340	585,000	.010062	3,886.27	5,850.00	— 36.27
3,176.9	2,370	592,500	.009989	5,918.48	5,925.00	+ 6.52
3,217.2	2,400	600,000	.009916	5,949.60	6,000.00	+ 50.40
3,257.4	2,430	607,500	.009843	5,979.62	6,075.00	+ 95.38
3,297.6	2,460	615,000	.00977	6,008.55	6,150.00	+ 141.45
3,337.8	2,490	622,500	.009697	6,036.38	6,225.00	+ 188.62
3,378.0	2,520	630,000	.009624	6,063.12	6,300.00	+ 236.88
3,418.2	2,550	637,500	.0095533$\frac{1}{3}$	6,090.25	6,056.25	— 34.00
3,458.4	2,580	645,000	.0094826$\frac{2}{3}$	6,116.32	6,127.50	+ 11.18
3,498.6	2,610	652,500	.009412	6,141.33	6,198.75	+ 57.42
3,538.8	2,640	660,000	.0093413$\frac{1}{3}$	6,165.28	6,270.00	+ 104.72
3,579.1	2,670	667,500	.0092706$\frac{2}{3}$	6,188.17	6,341.25	+ 153.08
3,619.3	2,700	675,000	.0092	6,210.00	6,412.50	+ 202.50
		Total...............		$113,333.20	$113,726.25	— $393.05

37. Referring to the tabulation, one may observe that between 2,190 and 2,370 kilowatts, the chart gives figures higher than the schedule, while from 2,370 to 2,520 kilowatts the monthly bills are lower. As previously stated above, the average is practically the same. The last line of the table shows that for the points taken the discrepancy between the chart and the schedule amounts to about $\frac{347}{1000}$ of 1 per cent. This difference would be lessened by choosing smaller sub-divisions and would finally amount to zero when the points are taken at 1 kilowatt intervals.

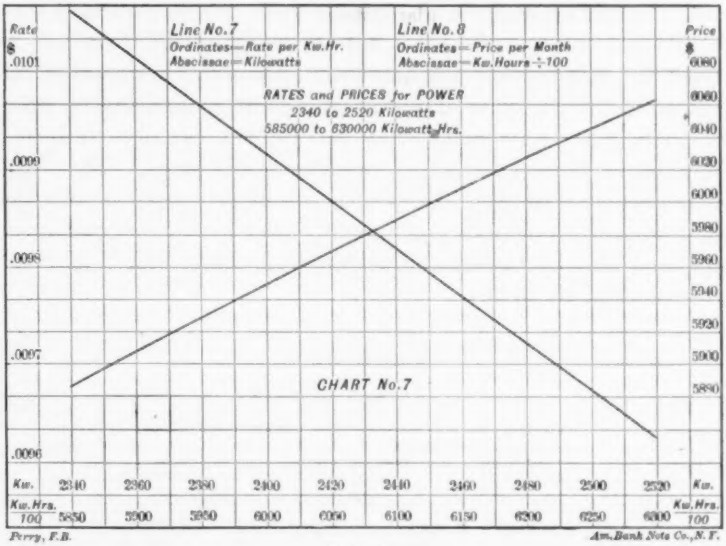

Rate $
Line No. 7
Ordinates = Rate per Kw.Hr.
Abscissae = Kilowatts
Line No. 8
Ordinates = Price per Month
Abscissae = Kw.Hours ÷ 100
Price $
RATES and PRICES for POWER
2340 to 2520 Kilowatts
585000 to 630000 Kilowatt Hrs.
CHART No. 7
.0101
.0099
.0098
.0097
.0096
6080
6060
6040
6020
6000
5980
5960
5940
5920
5900
5880
Kw. 2340 2360 2380 2400 2420 2440 2460 2480 2500 2520 Kw.
Kw. Hrs. / 100 5850 5900 5950 6000 6050 6100 6150 6200 6250 6300 Kw. Hrs. / 100
Perry, F.B.
Am. Bank Note Co., N.Y.

FIG. 31.

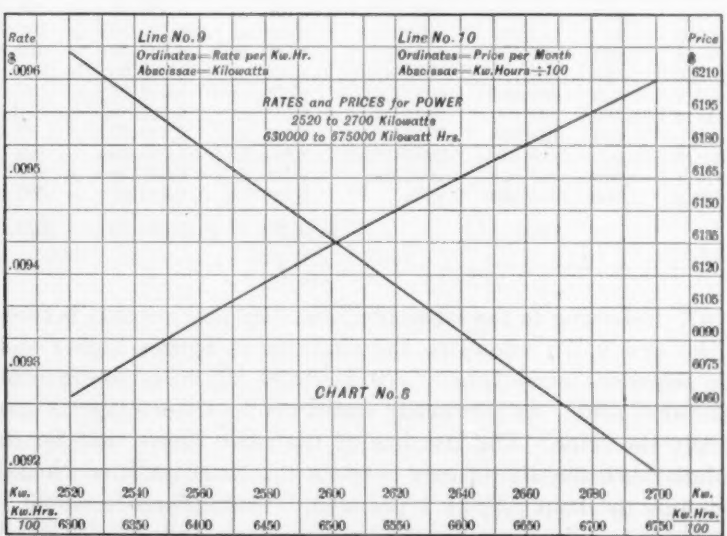

Rate $
Line No. 9
Ordinates = Rate per Kw.Hr.
Abscissae = Kilowatts
Line No. 10
Ordinates = Price per Month
Abscissae = Kw.Hours ÷ 100
Price $
RATES and PRICES for POWER
2520 to 2700 Kilowatts
630000 to 675000 Kilowatt Hrs.
CHART No. 8
.0096
.0095
.0094
.0093
.0092
6210
6195
6180
6165
6150
6135
6120
6105
6090
6075
6060
Kw. 2520 2540 2560 2580 2600 2620 2640 2660 2680 2700 Kw.
Kw. Hrs. / 100 6300 6350 6400 6450 6500 6550 6600 6650 6700 6750 Kw. Hrs. / 100
Perry, F.B.
Am. Bank Note Co., N.Y.

FIG. 32.

Of course, if the meter should indicate a use—say of 2,520 kilowatts per month considerably more would be paid to the central station by the chart system than by the schedule. It is equally true that there are points on the chart lines where, if the meter should read the proper amount, the bill would be less than by the schedule.

38. The convenience of this method in determining the value of bills rendered each month should be appreciated both by the electric company and by the user of current. There is no conflicting of rates or prices with such a system, and bills increase gradually and consistently, as they ought, in proportion to the amount of power used. This method, which is equally fair to producer and to consumer has, to the best of my knowledge, not been used up to the present time by any Central Electric Station for finding costs of current where varying rates per kilowatt annum or per kilowatt hour are involved.

39. These ideas are submitted with the belief that their adoption will bring about more satisfactory results than can ever be realized by a continuance of the so-called "Step" system of rates.

DISCUSSION.

Mr. R. S. Hale.—Mr. Perry's illustration is nothing more or less than the usual scheme of discount based upon quantity, although it is stated in figures per kilowatts. That is, he might set a price of $31 per kilowatt up to $60,000 per year and there would be no discount; above that there would be 5 per cent. discount, and so on. The bad feature of this plan is that it permits the use of a little more current, in some cases, actually to reduce the bill, which is taken care of in most electric companies by providing that the customer need in no case pay more than if he had actually used the amount necessary to obtain the higher discount. This, of course, is equivalent to making no charge for some of the current used. In other cases—as, for instance, the Chicago Edison Company—the contract provides, "Intermediate discounts to be determined by interpolation"; and this gives, I understand, no trouble in practice. It is of course practically just the method that Mr. Perry suggests. The best way, however, is what is known as the block system: By selling the first 2,000 kilowatts at say $41 each; the next 700 kilowatts at $29 each, making, how-

ever, no reduction on the first 2,000; the next 200 might be sold at $28 each, and so on. The proper average rate can be obtained in this way.

The most important feature, however, in determining rates is entirely omitted by Mr. Perry, and depends on the difference between kilowatts and kilowatts hours. Thus 2,000 kilowatts used for 3,000 hours is six million kilowatts hours, and a price of $30 per kilowatt is one cent per kilowatt hour. If now the amount should be determined by a kilowatt hour meter, the customer might use the six million kilowatt hours during only 1,500 hours of the year, using 4,000 kilowatts during that time; the kilowatt hour meter would still show six million kilowatt hours as before, but the central station would have to have twice the investment in engines, dynamos, boilers and so forth, that it would have to have if the current was used during 3,000 hours, and the central station would lose money. If, on the other hand, the customer used the six million kilowatt hours during 20 hours per day, or 6,000 hours per year, using day and night service, the central station would require only 1,000 kilowatts of machinery, only half the investment and outlay, and would make a much greater profit.

There are a great many systems in use by central stations for taking care not only of the feature that makes the argument for Mr. Perry's paper, but other important features that must be considered when a central station makes its rates. At Niagara Falls the published rates were $1 per kilowatt, based on the maximum taken, or the minimum power, that the customer used at any one time, corresponding to the amount of machinery the company had to keep ready for that customer, and, in addition, 2 cents per kilowatt hour for the current used up to 1,000 kilowatt hours, $1\frac{1}{2}$ cent per hour for the next 1,000 kilowatts hours, $1\frac{2}{10}$ of a cent for the next 1,000 kilowatts hours, and so on. These illustrate only one of the many methods that have been used in determining the charges for electric light and power.

*Prof. W. W. Crosby.**—At Woburn, Mass., a system was devised by Mr. L. R. Wallis, at that time General Manager of the electric light plant, called by him "The Foresee (4-C) System of Charging," which is interesting. In this system each customer paid certain union charges based, first, on capacity demanded and, second, current used. The name was suggested from the follow-

* Submitted after adjournment.

ing sentence: "A Capacity Charge and a Current Charge." The capacity charge was figured as follows:

F = Fixed charges per annum on unit cost of plant.

N = Number of sixteen-candle-power lamps per unit.

C = Average "capacity charge" per annum per sixteen-candle-power lamp.

$$\frac{F}{N} = C.$$

The following quotation from the paper delivered by Mr. Wallis before the National Electric Light Association in 1901 explains the system further:

"After obtaining the average capacity charge per sixteen-candle-power lamp, the minimum and maximum charge per lamp is easily determined, and a sliding scale between the minimum and maximum is readily made that will result in an equitable distribution of the fixed charges among the various customers. While it is perfectly consistent to expect the customer to reimburse the station for all the fixed charges that it has to meet to furnish him with the service he requires, the proposition is only to insure the station against loss in carrying capacity for him, and to secure this protection it is not necessary to charge the full amount shown as being the total fixed charges, as the object is accomplished when a large proportion of the fixed charges are guaranteed.

"The probable fact that all of the consumers will not demand their maximum contracted capacity at the same time should be taken into consideration in establishing the capacity tariff, but should not be made the basis for individual concessions."

The capacity schedule is made up on the basis of the number of lamps demanded on a yearly rate and a monthly rate. There is also a schedule of rates from October to June, and then for June to October, this latter charge being less than the charge for winter months. I may add that in the practical working of the system I found that as a customer there was much to commend it.

*Mr. Perry.**—Replying to Mr. Hale's remarks, I wish to say that the contract, which contained rates similar to those chosen for the illustration in the paper, included no clause of any kind which would produce a discount or reduction in bills other than that covered by the schedule itself. The omission from the contract of the usual proviso, which stipulates that "the customer

* **Author's Closure under the Rules.**

shall not be billed a less amount for the use of a larger quantity of current, alluded to by Mr. Hale, has in this case served to bring out even more strongly the existing fault in the present "discount system." With the latter it is a fact, as pointed out by Mr. Hale, that no charge is made for some of the current used. There is, therefore, something radically wrong with the system on account of this inconsistency, which could not possibly occur with the "chart system."

To make the defect more apparent, the following figures are interesting, because they indicate what ordinarily happens in applying the "discount system." This data was given me by a gentleman who formerly operated a 5 horse-power, 500 volt, direct current motor from a central station circuit. The minimum charge per month in this instance was $3, and the base rate was 10 cents per horse-power hour. Here are some of the bills:

(1) April 1st to April 24th,	211	horse-power	hours,	discount	30	p. c.	$14.77
(2) " 24th " May 27th,	310	"	"	"	40	"	18.60
(3) May 27th " June 25th,	192	"	"	"	20	"	15.36
(4) June 25th " July 28th,	247	"	"	"	30	"	17.29
5) July 28th " Aug. 27th,	194	"	"	"	20	"	15.52

Comparing bill No. 1 with bill No. 5, it is noticeable that a current consumption equivalent to 17 horse-power hours less was used in the latter case, although the total cost for so doing was 75 cents more than in the first instance. Item No. 2 indicates that during this period about 25 per cent. more current was consumed than in the time covered by bill No. 4, although the increase in cost was approximately only 8 per cent. The quantities of current considered in the paper are much larger, consequently the inconsistencies are more marked; the principle, however, is the same in both cases.

The title of the paper may be misinterpreted, since it is misleading; the text indicates, however, that it is not intended to suggest a method for establishing the original basis for the rates. On the assumption that these have been fixed, the paper illustrates, by means of charts, a method for determining rates per kilowatt hour and prices per month for electric current. No attempt has been made to define what elements should be considered in finding the cost of central station operation for the purpose of properly adjusting the rates. These vary with local conditions and must be carefully worked out in every individual case. Assuming the cost to have been found in any specific case

for producing electric current in small quantities, as well as for successively larger amounts up to the total normal capacity of the plant, I maintain that rates may be arranged consistently in proportion to the current output without the use of the discount sheet. This result may be brought about by the "chart system," which, when once applied to meet the conditions of any plant, is free from the objectionable features now existing. In most instances the rates would be expressed by curved lines instead of by straight lines as shown in Figs. 25 to 32 inclusive. In order to avoid confusion, it would possibly be better to omit the rate curve from the chart issued to the customer, since that of greater importance is the one from which monthly bills may be determined at a glance.

It is customary in nearly all contracts to insert a clause which specifically states the minimum monthly charge which the customer is obliged to pay to the central station. In view of this fact, and since all recording wattmeters indicate kilowatt hours, it is evident that a central station would have to make a special contract covering the unusual conditions of operation discussed by Mr. Hale. The subject considered is not due to any misapprehension of facts or to a lack of knowledge of the usual forms of contracts made by central stations. The case discussed was not assumed, but was met with in actual practice, although, as stated above, the rates and quantities were slightly altered for obvious reasons. It may be that I have been presumptuous in calling attention to inconsistencies of a system which is almost universally followed and with which every one is more or less familiar. The fact that this system is acknowledged to be faulty in practical operation is sufficient reason for its displacement if some better and more coherent method may be devised. The discussion which the paper has merited has borne out my contention that the "discount system" is an unfair one, consequently is open to improvement. Graphical or diagrammatic representation has long since been recognized as giving solutions quickly and nearly, if not quite, as accurately as methods involving mathematical computation, and I am yet to be convinced that this system is unworthy of consideration as a satisfactory substitute for the "discount" or "step system."

NOTE.—The original charts were made on laboratory cross section paper containing ten more spaces horizontally and vertically than the squares reproduced in Figs. 25 to 32 inclusive. Paragraphs 30, 32 and 35 should be interpreted with this understanding in mind; otherwise it will appear that the text is incorrect.

No. 1017.*

AN IMPROVEMENT IN VALVE-MOTION OF DUPLEX AIR COMPRESSORS.†

BY STERLING H. BUNNELL, LORAIN, OHIO.

(Junior Member of the Society.)

1. The use of poppet valves held down by springs operates in pumps handling incompressible fluids merely to increase the work done by the piston, generally only by a trifling amount. The same system of valves applied to a compressor working with an elastic fluid not only involves a similar loss by the friction of the fluid in passing the spring-loaded valves, but also decreases the density of the fluid filling the cylinder at each stroke, so that the total weight of gas handled falls considerably below that corresponding to the swept capacity of the cylinder. Such valves by reason of their inertia tend also to delay closing till after the reversal of the motion of the piston at the end of its stroke and thus to cause a further loss by slippage. For these reasons mechanically-actuated inlet valves are generally applied to compressors of medium and large size. The adaptability of the duplex or two-crank type of direct-connected air-compressor to varying capacity requirements and occasional unusually low speed, together with the superior economy of its steam cylinders working under short cut-offs, has brought about its general use except for supplying such small quantities of air as can be delivered by the single cylinder of the "straight-line" or single crank tandem compressor of small size. It is to the common type of duplex compressor with

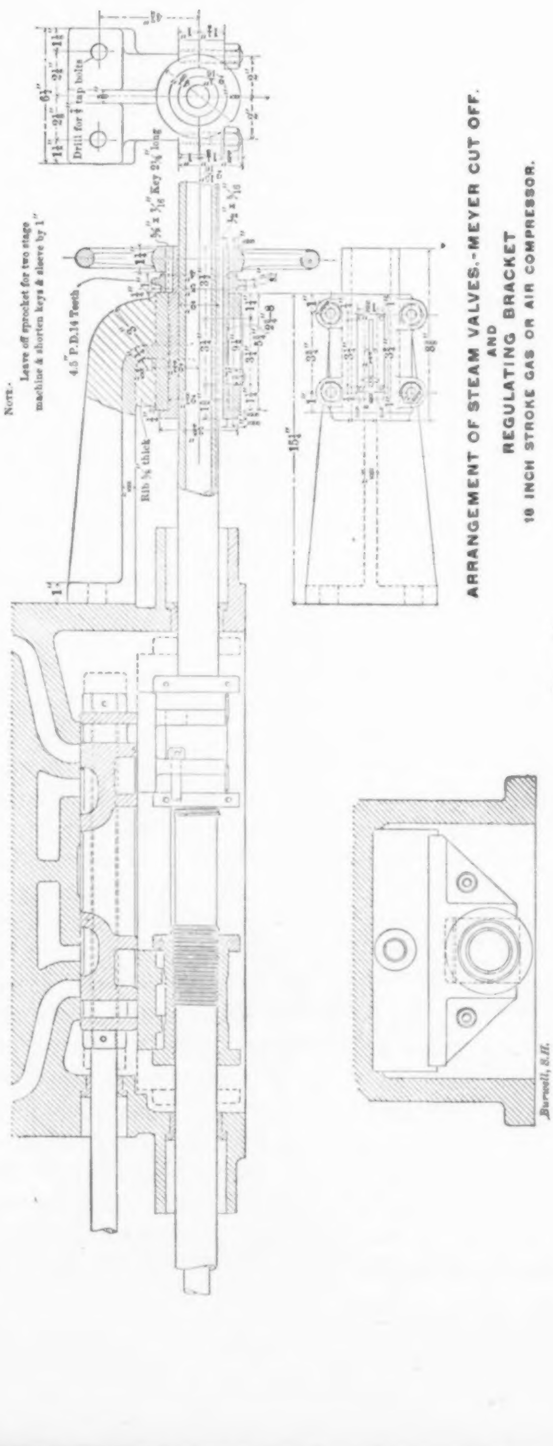
---

* Presented at the New York meeting, December, 1903, of the American Society of Mechanical Engineers, and forming part of Volume XXV. of the *Transactions*.

† For further discussion on this topic, consult *Transactions* as follows:

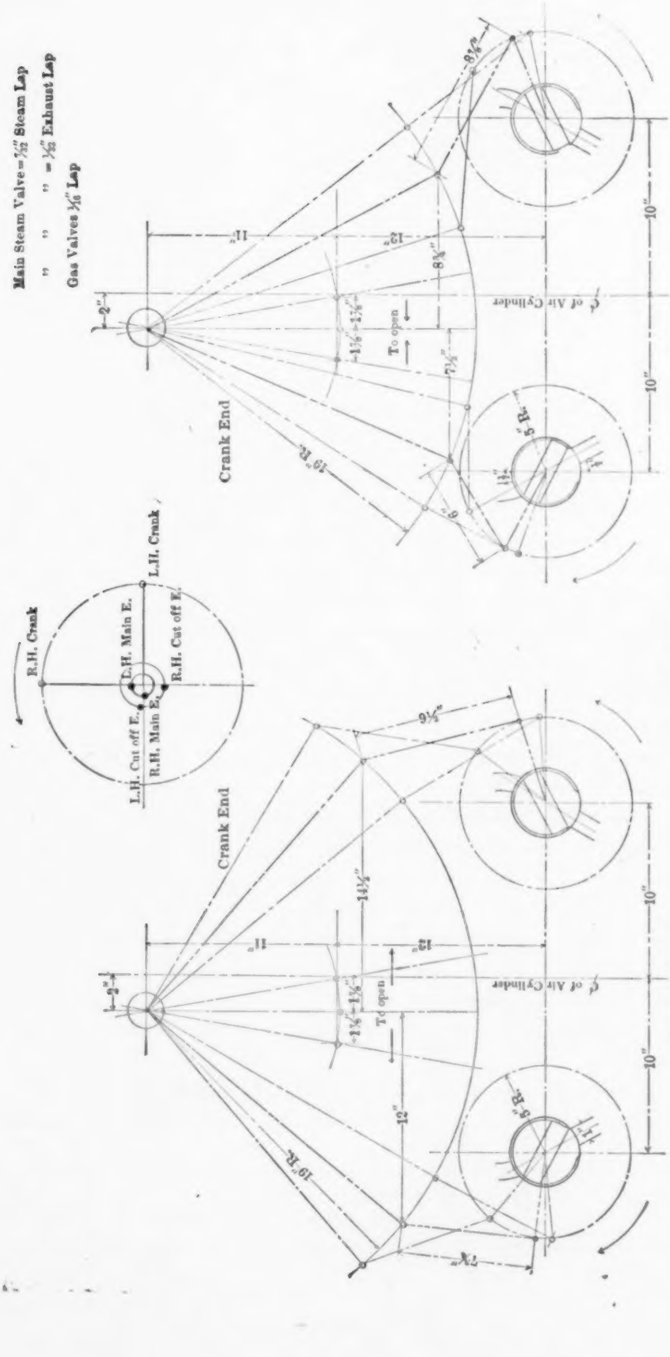
No. 824, Vol. xx., p. 967: "New System of Valves for Steam Engines, Air Engines, and Compressors." E. W. Gordon.

No. 920, Vol. xxiii., p. 151: "New Valve Gear for Gas, Steam, and Air Engines." E. W. Naylor.



ARRANGEMENT OF STEAM VALVES.-MEYER CUT OFF.
AND
REGULATING BRACKET
18 INCH STROKE GAS OR AIR COMPRESSOR.

Fig. 33.



VALVE DIAGRAM
GAS CYL'S 13"X 18"HOR. DUP'X GAS COMP.

FIG. 34.

Meyer steam cut-off valve gear and mechanically-actuated air-inlet valves that the construction to be described applies.

2. The main steam valve of a Meyer or riding cut-off gear is set exactly like any plain slide-valve, being laid out to give proper steam lead, exhaust opening and exhaust closure or compression, without regard to the point of cut-off, which will therefore come somewhere around $\frac{3}{4}$ or $\frac{7}{8}$ full stroke. The riding cut-off valve is operated by an eccentric set either just opposite the crank, or better, a little back of this position. The air-inlet valves are usually of rotary type, driven like Corliss steam valves by a rocker or wrist-plate connected to the valve-arms by short links, or they may be plain slide valves. In any case the inlet valve must close as the piston reaches the end of its stroke, and open shortly after it commences the suction stroke, the lateness of opening being for the purpose of allowing the air contained in the clearance space to expand to the pressure of the air in the intake. The required position of the eccentric operating air-inlet valves is therefore approximately at right angles to the crank operating the piston, or more or less back of this position.

3. Air-compressors of the type just described have been regularly provided with six eccentrics to operate the double steam valves and air-inlet valves of the two sides of the duplex machine. A moment's consideration of the preceding paragraph will show that the cut-off valve eccentric of one steam cylinder and the air-inlet valve eccentric of the opposite cylinder are a little back of the position opposite the crank on the steam cylinder side, while the alternate steam and air cylinders have valves to be actuated by eccentrics, one set back of the position opposite the crank and the other 180 degrees from the first. It is only necessary to modify the arrangement of valve-arms and links of the inlet valves of the latter compressing cylinder, using precisely similar valves, valve-arms and other gear except the rockers or wrist-plates, to allow of driving the air-valve gear of each side of the machine by direct connection to the cut-off valve rod of the other side.

4. The details of the gear are clearly shown by the illustrations. Steel tube is common enough to-day to allow of using it for the right-and-left threaded sleeves of the cut-off valves. Through each sleeve is passed a smaller solid rod connected at one end by means of suitable rockers and pins to the eccentric rod, and at the other end to an arm carried on a cross-shaft. This rod drives the encircling sleeve, and, therefore, the cut-off valves carried on it, through the medium of two simple split clamps touching

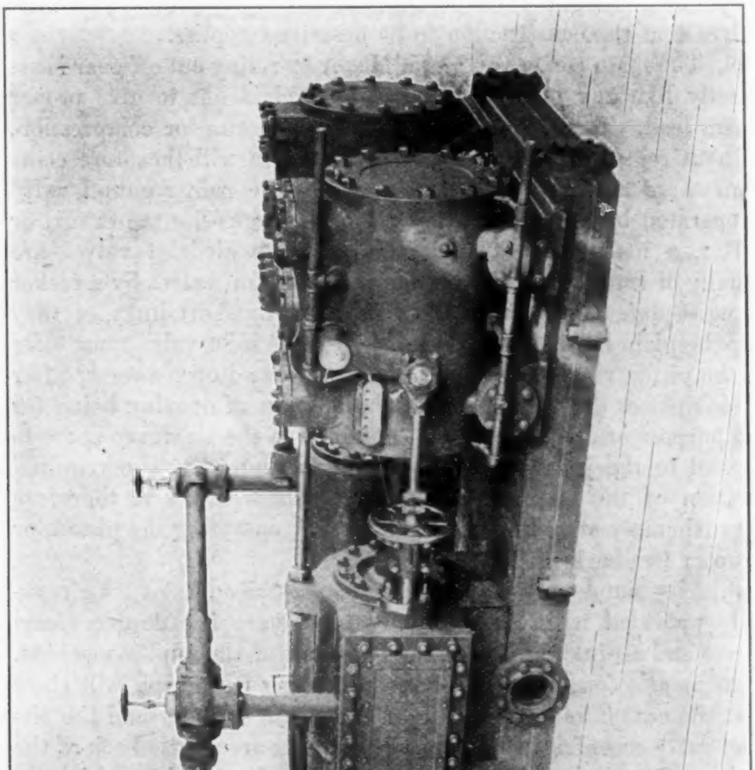


FIG. 35.

the ends of the sleeve. The cross-shaft passes through bosses on the compressing cylinder casting and terminates in a boss on the opposite compressing cylinder, and carries adjacent to the latter boss the wrist-plate or rocker driving the inlet valves of this side. This description applies also to the details of the corresponding gear of the other two cylinders. It happens that there is absolutely no difference between the two sets of parts except in the two wrist-plate rockers, and in the fact that one of the long rods is a little longer than the other because the centre of its rocker is farther back on the air cylinders.

5. It looks at first sight as if this combination would be troublesome to lay out and to adjust in erecting the compressor. There is really no additional complication, for it is only necessary to slack off the clamps on the long rods, set up the air-valve motion in the usual way, locate the eccentrics as required and then set the sleeve along the rod to give even cut-offs and tighten the collars. The cut-off valves can be changed to equalize the cut-offs at any time without disturbing the long rod, merely shifting the clamps as desired, and the air valves may be shifted or reset without disturbing the equality of the cut-offs. The net result of the arrangement is the saving of two eccentrics, straps, rods and rockers and of the space between or outside the cylinders that would otherwise be occupied by these parts, at the cost of substituting a piece of steel tube for a solid rod, and of enlarging the diameters of steam chest glands to correspond. The feature of operating one side of the machine alone in case of necessity is not lost, because as long as the crank-shaft is not broken and the cylinder castings remain in position, the eccentric driving the air-inlet valves of the cylinder which is to be operated may as well be on one end of the shaft as on the other.

6. The compressor shown embodies a modification of the usual framing, which has some advantages. The two separate bed plates are bolted together along their centre lines, and are further bolted to a single cross member lying under the cylinders. The strains which tend to work the ordinary duplex machine on its foundations are thus resisted directly by the cross frame, making the machine nearly independent of a masonry foundation except as a mere support. Tie rods between the upper parts of cylinders and over the guides add greatly to the rigidity of the whole.

7. A number of these machines have been constructed ranging from 12 to 24 inches stroke, all of which have been shipped without taking apart and leveled upon foundations in a few minutes'

time. Some of these compressors have been continuously operated at speeds up to 200 revolutions per minute for months together, and none of them have developed any objection to the combination of valve mechanism described, or have shown the tendency of the usual duplex compressor with independent frames to shift on its foundations and thus work out of alignment.

No. 1018.*

TESTS OF A DIRECT CONNECTED EIGHT-FOOT FAN AND ENGINE.†

BY E. S. FARWELL, NEW YORK, N. Y.
(Member of the Society.)

1. THE promoters of the plenum system of heating and ventilating have obtained a very substantial footing among the paper mills, because of the moisture which it is necessary to remove from the rooms.

For instance, in a room containing two paper machines, each making 25 tons of paper in 24 hours, there is approximately 100 tons of water which must be disposed of before it condenses on the roof and trusses, and drips on the machines. Also, in pulp-grinder rooms, there is a large amount of steam generated which must be disposed of, or the fog in the room will be so dense as to impede the work. In most of the other rooms, however, it is merely a question of heating.

I found it rather difficult to obtain reliable data which would guide me to an intelligent selection of apparatus for any particular job. The builders of such apparatus very generously offered to work out the problem for me, but it is safe to say that in no case were any two proposals for the same size of apparatus. "A" would submit a proposition for a small fan, running at high speed, with small air-pipes and a small heater; "B" would offer a large fan to run at a slow speed, with large air-pipes and a large heater; "C" would perhaps offer a large fan, with large air-pipes and a small heater. If "A" were asked to furnish a

---

* Presented at the New York meeting (December, 1903) of the American Society of Mechanical Engineers, and forming part of Volume XXV. of the *Transactions*.

† For further references on this subject, see *Transactions* as follows:
No. 240, vol. viii., p. 313: "Power to Drive Fans."
No. 264, vol. ix., p. 51.

large fan to run at a slower speed, he immediately proposed to enlarge the pipes proportional to the size of outlet of the fan, and to put in a much larger heater, notwithstanding the fact that the amount of heat to be delivered was the same. I wish to further acknowledge that under these perplexing difficulties the representatives of the different fan builders never hesitated to give me all the assistance which they apparently could, and I believe that I have finally succeeded in getting some points clear in my own mind.

2. The question seemed to me to embody three distinct problems: the air-pipe, the fan, and the heater. The first two are interdependent, or perhaps we had better say both depend upon the same assumed data. The starting-point of the problem or problems is the volume and temperature of air required. In a factory building, if it is merely a question of heat, a comparatively small volume of air at a high temperature is satisfactory. A change of air as low as 30 minutes is allowable where men do not stand at their work, and 20 or 25 minutes where they do. In the grinder-room the change must be as frequent as 15 or even 10 minutes, depending on the shape of the room, the location of the grinders, kind of roof, and other local conditions. In paper machine rooms, also depending on local conditions, the required change may be as often as 4 minutes.

It is very easy, with the published tables and data of tests, to determine the amount of heating surface required to heat the determined volume of air to the desired temperature. I never could see how the size of the fan need affect the amount of heating surface, provided the volume to be delivered and initial temperature of air were the same in both cases.

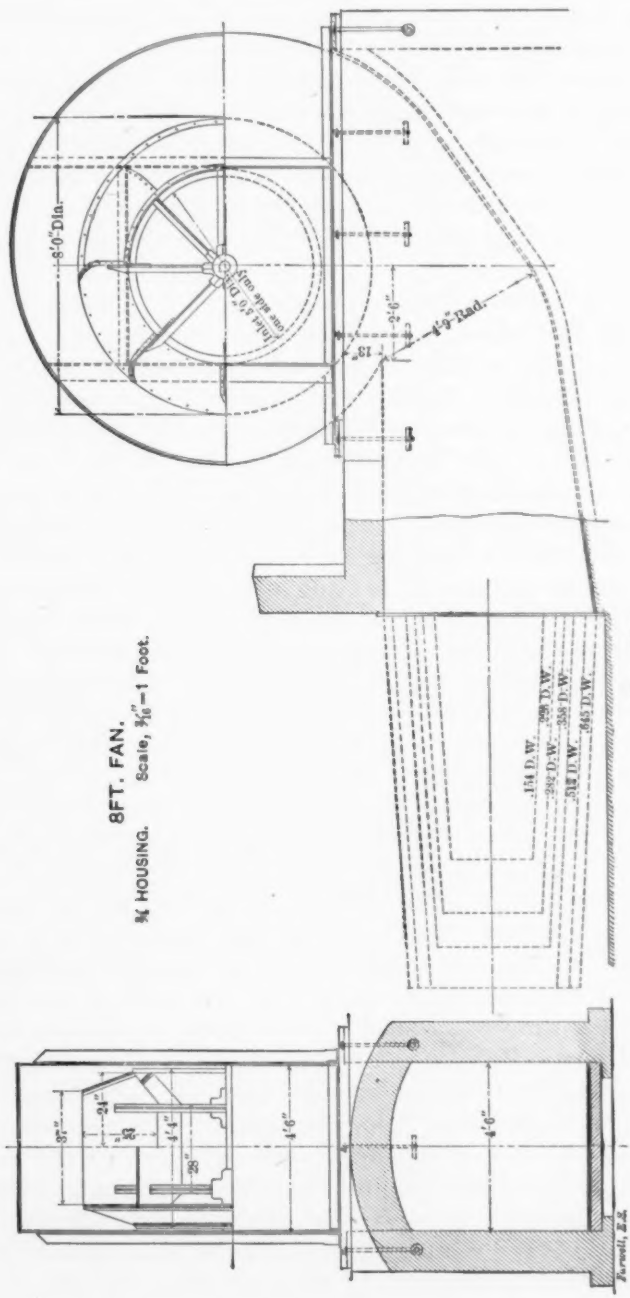
3. About $\frac{1}{16}$ ounce pressure per square inch will be required to give the necessary velocity at the outlets of the distributing pipes. In addition to this there must be as much more pressure as is necessary to overcome the friction of the pipes, and herein I find is the principal difference in the practices of the different builders. Some always figure on 1 ounce pressure at the fan. Others figure $\frac{5}{8}$ ounce. Fifteen-sixteenths of an ounce seemed to me to be an excessive loss, and I usually call for $\frac{1}{2}$ ounce at the fan, enlarging the pipe sufficiently to bring the friction loss well within that figure.

In reducing the pressure required of the fan, we have of course reduced the amount of work done by the fan; but against this

saving in work, it is necessary to charge the interest and depreciation on the increased cost of fan and pipe.

4. In order to secure some data for my guidance, I made a series of experiments on the efficiency of a fan which I purpose to outline in this paper. The fan was a No. 160, according to the usual method of designating fan sizes, with a wheel 8 feet in diameter, and 37 inches wide at the periphery and with one side inlet. It is shown in Fig. 36. The fan is driven by a direct-connected steam engine, and discharges into a large chamber supplying air for combustion to the boilers. This chamber was left open to maintain atmospheric pressure except in a couple of tests. The opening of the fan into this chamber was tightly boarded up, and conical tubes or nozzles were fastened to circular openings in this board partition. The tubes had a taper of 3½ degrees and, with the exception of the largest two sizes, were approximately three times the diameter in length. Six different sizes of tubes were tested, each at 8 or 10 different speeds, ranging from 50 to 250 revolutions per minute. In addition, a set of observations at each speed was made, with no outlet whatever.

5. This I realize is not an ideal arrangement, but under the circumstances seemed to be all that the desired results would warrant. The volume of air delivered was determined by means of a pitot tube located at the extremity of the conical outlet. We attempted to use an anemometer, but some of the velocities were beyond the capacity of the instrument and it very soon proved to be unreliable. With the tapering outlet, it was apparently safe to assume that the air at the extremity had no static pressure, but that the entire potential energy had been converted into kinetic energy. The pressure in the fan chamber, as also the vacuum or suction at the inlet, was measured by a water column. We found the vacuum at inlet varied at the different points of the inlet as the gauge was moved from the centre toward the rim. A number of observations were made, and a point determined which gave us the mean of all the readings. The temperature of the air was taken, also the revolutions of the fan and indicator cards from the engine. This completed the list of observations taken during each test. Each test was run for half an hour under uniform conditions, and readings taken every five minutes. The average results of each test are given in the table. The volume of air discharged was computed from the velocity of air as given by the pitot tube, and tabulated in



8FT. FAN.
¾ HOUSING. Scale, 3/16"=1 Foot.

FIG. 33.

column marked "B." The theoretical horse-power was computed from the volume, determined as above noted, and the pressure in chamber as tabulated. The pressure, volume, indicated horse-power, and efficiency have also been plotted on the two sets of curves, Figs. 37 and 38. In Fig. 37 the abscissæ represent revolutions of fan, while each curve represents a certain size of outlet. In Fig. 38 these two items have been interchanged, and each pressure curve has a different zero line. It is the custom, I believe, to designate the size of fan outlet in terms of the diameter and peripheral width of fan-wheel, and this I have done in the present instance.

6. A number of interesting facts stand forth very clearly upon an examination of these curves. Theoretically, the volume should vary directly as the speed of the fan with a given size of outlet, the pressure as the square of the speed, and the horse-power as the cube of the speed. The curves show that the ratios are not quite those stated. Up to a certain point the volume curves are very nearly straight, but at the higher speeds they seem to fall off. This falling off was more marked with the large openings, as was also a large increase of the vacuum at inlet. This loss in the volume delivered is unquestionably due to the throttling of the air at inlet. The maximum efficiency of the combined unit was secured at 142 revolutions per minute, when the pressure was $\frac{1}{2}$ ounce (this, of course, only applies to the particular fan tested, and it is fair to presume that if the fan had had an inlet on each side, the throttling would have been less and the most efficient speed might have been higher).

7. The curves show very clearly, however, that in the selection of fan we should choose large sizes running at moderate speeds and developing a low pressure. We may apparently run the same fan at a lower speed and lower pressure so that it will deliver the same volume of air with a considerable saving in horse-power. This should be done, if in any case the conditions prevent the selection of a larger fan. As I will illustrate in a few moments, however, the larger fan will do the work more efficiently.

It has been frequently stated that up to a certain size of outlet, variously styled the "theoretical outlet," "square inches of blast," and "capacity of wheel," any change in size of outlet makes no change in the pressure, and a variation in volume and horse-power directly proportional to the size of the outlet, and

that further enlarging results in a decided drop of pressure and falling off in the rate of increase of volume and horse-power.

The curves show that up to about .29 D. W. the pressure drops

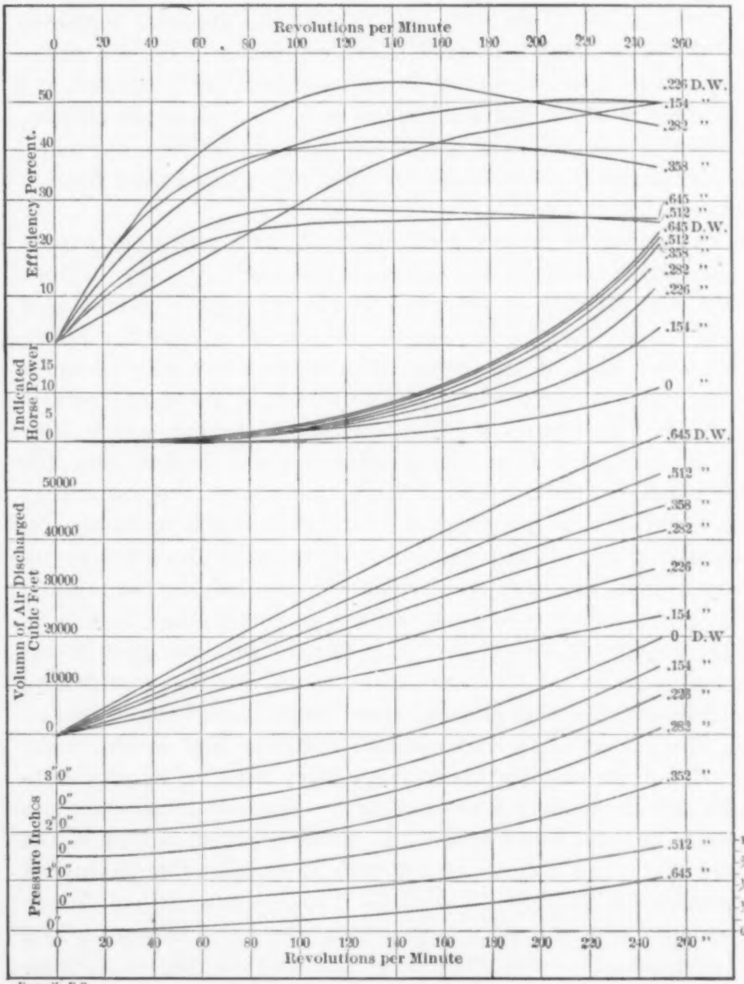


FIG. 37.

but slightly. The volume and indicated horse-power rise very nearly by straight lines, but beyond that point there is a sudden and rapid change, the efficiency also reaching a maximum at this same point.

8. As being probably pertinent to this point, I wish to call attention to the fact that the area of the fan-blade is equal to .2885 D. W. The efficiency curves show that but slight varia-

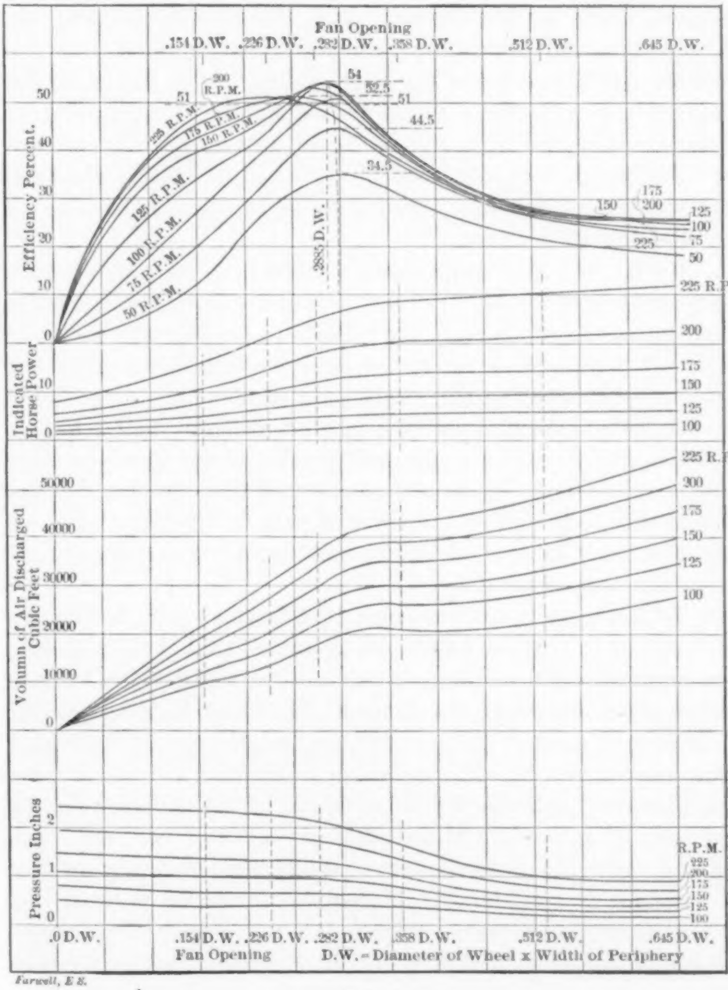


FIG. 38.

tions from this "theoretical outlet" should be permitted. Mr. Snow states that for general practice the square inches of blast is not far from $\frac{D.\ W.}{3}$. I understand that the usual width of pe-

riphery of an 8-foot fan, as built by the makers of the fan tested, is 41 inches instead of 37 inches. This would make the area of fan-blades approximately .32 D. W., which might increase the effective area, or "theoretical outlet," to a like quantity and bear out the statement made by Mr. Snow. This "theoretical outlet," as has been stated by Mr. Snow, is not to be understood to be the actual sizes of the outlet of the fan's casing, but to be the size of opening which will offer a resistance equivalent to the sum of all resistances of distributing pipes. If in any given case we are able to state what this equivalent outlet is to be, we can then select the size of fan which will give us the desired volume of air at the desired pressure.

9. In Fig. 39 I have reproduced the efficiency curves and have drawn another curve showing the relation existing between the size of outlet and the ratio of air velocity to peripheral velocity of wheel. It is not safe to reason too much from the concrete to the abstract, but comparing this curve with similar ones constructed from the tests of a small pressure blower, as given in Mr. Kent's handbook, and from the guaranteed results of one blower-maker on his standard fans (presumably computed from tests which are not available to the public), I think it safe to say that the curve of the tests may be applied to different sizes of fans with reasonably satisfactory results.

10. To illustrate the advantages that have already been mentioned, we have only to notice, for instance, that the fan tested, when running at 200 revolutions per minute, will develop 1 ounce pressure when delivering a volume of 32,000 cubic feet per minute, at an efficiency of 51 per cent. and requiring 17.2 indicated horse-power to run it. The same fan running at 161 revolutions per minute will deliver the same volume of 32,000 cubic feet at a pressure of ½ ounce and an efficiency of 41.7 per cent., but requiring only 10½ horse-power. While the efficiency in the first instance is considerably higher, it should be pointed out that the actual horse-power required is 6.7 more. This represents the actual cost of delivering 32,000 cubic feet of air into the room under the two pressures noted. Again, this fan running at 142 revolutions per minute will deliver 25,800 cubic feet of air at a pressure of ½ ounce, and at an efficiency of 54 per cent. If, however, it is desired to deliver 32,000 cubic feet of air at ½ ounce pressure and at maximum efficiency, it will be necessary to use a larger size of fan, the determination

of which will illustrate the use which I have made of these tests.

I am not prepared to say that for fans with inlet on one side

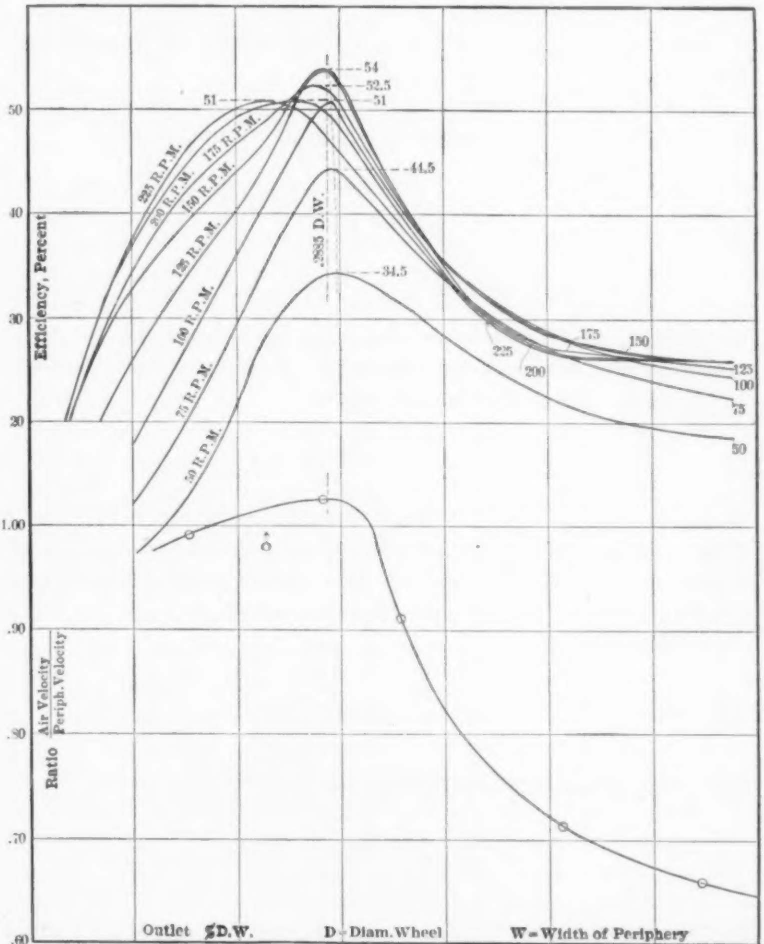


FIG. 39.

only, .29 D.W. is as large a "theoretical outlet" as should be used. However, it seems to be evident that where inlet may be had on both sides, a wider fan may be used, and possibly further experiments may show that a wider fan may be used with a single

inlet. Assuming that .29 D.W. is to be used, we have for the volume of air delivered,

$$Q = .29\ D.\ W.\ V,$$

in which V is the air velocity corresponding to the assumed pressure.

11. The velocity corresponding to the pressure of $\frac{1}{2}$ ounce is 3653.8. From the foregoing equation we find that,

$$D.\ W = \frac{32000}{3653.8 \times .29} = 30.2.$$

If $W = \frac{D}{r}$, then $D^2 = 30.2\ r$. The proper value of "r" was not determinable from these tests, but probably has been determined by the builders' experiments. It is apparently between 2.3 and 2.7. In the fan under test,

$$r = 2.595.$$

It is desirable to use standard diameters and vary the width within certain limits. In the present case the size of fan selected should be $8\frac{1}{2}$ feet diameter by 42.6 inches wide at periphery. For the speed of fan, we have from the lower curve in Fig. 39 for .29 D.W.,

$$\frac{V}{V_p} = 1.025.$$

From this equation the peripheral velocity,

$$V_p = \frac{3653.8}{1.025} = 3570 \text{ ft. per min.}$$

This is equivalent to 133.5 revolutions per minute of an $8\frac{1}{2}$-foot fan. By the efficiency of 54 per cent. shown by the chart, this would require 8.125 horse-power, a still further saving over the 8-foot fan running 161 revolutions per minute.

12. There is another problem upon which I should be glad to have discussion and enlightenment. In designing an induced draught plant some time since, I proposed to install two fans, which, under normal conditions, would both run at slow speed,

either of which in an emergency could be speeded up to do the work of both. One representative informed me that he did not think it could be done practically, and after I insisted that it be worked out, two 10-foot fans, 24 inches wide at the periphery, were offered. After some discussion and computations, two 10-foot fans 54 inches wide at periphery were purchased. I based my contentions on the tests which I have described, although I am aware that the conditions when handling gases at a high temperature are somewhat different. I do not see, however, that these changed conditions prevent the use of a fan in the way I described. To illustrate, the following data may be taken from the curves of the 8-foot fan tested. With an outlet of .265 D.W., and running at 150 revolutions per minute, the fan will deliver 25,000 cubic feet of air at a pressure of $\frac{9}{16}$ ounce, requiring 7.8 horse-power, and give an efficiency of 53 per cent. This same fan with an outlet .53 D.W., running at 225 revolutions per minute, will deliver 50,000 cubic feet of air at the same pressure of $\frac{9}{16}$ ounce. In the latter case the horse-power is 40.8, and efficiency but 26.5 per cent. This efficiency is very low, but is not to be considered in an emergency. The only question which requires particular attention is an arrangement which will allow a change of the outlet, or its equivalent resistance, from .265 D.W. to .53 D.W., and this was very easily accomplished in the case mentioned. It is necessary in many industrial plants to provide against a shut down, due to any ordinary accident, but I fail to see the necessity of putting in two fans, each sufficiently large to do the entire work under normal conditions, at maximum efficiency.

I have not entered into this discussion with any feeling akin to the old Quaker's who is reported to have told his wife that "all the world is queer except thee and me, and thee is a little queer." But I have hoped to provoke a discussion by those who have had better opportunities for studying the problem than I have, and can give us information not obtainable, at least convincingly, from these tests.

In closing I wish to acknowledge the able assistance and painstaking care of Mr. C. W. Wilder, who made most of the observations and computations for me. But for my confidence in his ability, I should have hesitated to present these results to the Society.

TEST OF 8-FOOT FAN.

DIAMETER OF WHEEL, 8 FEET; WIDTH OF PERIPHERY, 37 INCHES.

1	2	3	4		5		6		7		8	10	11	12	13	14
SIZE OF FAN OUTLET.	Speed, revolutions per min.	Peripheral velocity, feet per minute. *A*	Vacuum at inlet. *C*		Pressure in chamber. *D*		Velocity head at orifice, by Pitot tube. h = inches.		Total head. *C + D*		Velocity of air, $v = 60 \times 64.5\sqrt{h}$ ft. per min. *B*	*B* ÷ *A*	Cubic feet of air discharged per minute.	Theoretical horse-power.	Observed horse-power.	Efficiency per cent.
			Inches.	*Ounces.*	*Inches.*	*Ounces.*	*Inches.*	*Ounces.*	*Inches.*	*Ounces.*						
	44.2	1,109	.100	.057	.116	.066	.110	.063	.216	.123	1,284	1.15	4,868	.088	.330	26.6
	74.5	1,869	.111	.063	.217	.125	.191	.110	.328	.188	1,691	.90	6,411	.218	1.040	21.0
A	102.0	2,510	.120	.069	.397	.229	.362	.209	.517	.298	2,320	.92	8,831	.551	2.000	27.5
	137.3	3,445	.154	.088	.903	.516	.838	.482	1.057	.604	3,543	1.03	13,420	1.890	4.27	44.3
.154 D. W.	140.6	3,526	.156	.089	.858	.494	.832	.480	1.014	.583	3,530	1.02	13,380	1.805	4.92	36.7
26½ in. diam.	153.6	3,855	.237	.136	1.046	.600	.943	.543	1.283	.736	3,758	.97	14,240	2.335	5.91	39.5
3.79 sq. ft.	155.9	3,912	.280	.165	1.076	.615	1.047	.608	1.356	.780	3,960	1.01	15,010	2.515	5.98	42.1
	190.6	4,784	.281	.167	1.710	.984	1.650	.950	1.991	1.151	4,971	1.02	18,840	5.060	....	....
	192.6	4,834	.454	.230	1.562	.898	1.518	.868	2.016	1.158	4,768	.98	18,080	4.430	10.34	42.8
	239.6	6,013	.477	.275	2.668	1.513	2.620	1.500	3.145	1.788	6,264	1.04	23,750	9.810	20.07	48.9
	49.4	1,239	.054	.031	.131	.075	.117	.067	.185	.106	1,324	1.06	7,374	.151	.39	38.7
	66.8	1,676	.070	.040	.224	.129	.202	.116	.294	.169	1,749	1.04	9,741	.343	.76	45.2
	108.8	2,730	.142	.081	.487	.280	.451	.260	.629	.361	2,598	.95	14,490	1.107	2.71	40.8
B	109.5	2,774	.187	.107	.495	.285	.443	.255	.682	.392	2,586	.93	14,430	1.125	2.70	41.6
	120.5	3,024	.218	.126	.613	.352	.546	.316	.831	.478	2,860	.95	15,930	1.530	3.59	42.7
.226 D. W.	134.5	3,375	.250	.144	.734	.424	.692	.398	.984	.568	3,219	.95	17,930	2.075	4.78	43.4
32 in. diam.	158.2	3,970	.285	.164	1.210	.695	1.110	.640	1.495	.859	4,080	1.03	22,730	4.320	8.32	52.0
5.56 sq. ft.	161.2	4,046	.370	.213	1.057	.605	.957	.550	1.427	.818	3,786	.93	21,080	3.490	8.01	43.6
	173.4	4,347	.380	.219	1.441	830	1.299	.748	1.821	1.049	4,411	1.02	24,670	5.580	....	....
	189.6	4,750	.700	.403	1.375	.785	1.312	.755	2.075	1.188	4,431	.93	24,680	5.285	11.84	44.6
	202.8	5,090	.410	.236	2.000	1.145	1.811	1.045	2.410	1.381	5,208	1.02	29,420	9.060	16.21	55.8

C .282 D. W. 35¾ in. diam. 6.95 sq. ft.	51.1	1,275	.070	.040	.140	.085	.142	.081	.210	.125	1,458	1.14	10,160	.236	.71	33.3
	76.0	1,907	.100	.057	.281	.163	.282	.162	.381	.220	2,055	1.07	14,320	.637	1.35	47.25
	102.8	2,580	.192	.112	.476	.274	.478	.275	.668	.386	2,676	1.03	18,650	1.395	2.81	49.6
	115.0	2,886	.260	.149	.605	.347	.607	.350	.865	.496	3,015	1.04	20,980	1.990	3.82	52.1
	136.3	3,420	.293	.169	.816	.470	.813	.468	1.109	.639	3,496	1.02	24,330	3.122	5.53	56.5
	151.3	3,797	.280	.163	1.007	.577	1.004	.577	1.287	.740	3,877	1.02	27,020	4.260	7.99	53.3
	177.4	4,452	.480	.277	1.438	.825	1.428	.820	1.918	1.102	4,624	1.04	32,180	7.245	13.52	52.8
	187.8	4,713	.550	.317	1.479	.845	1.457	.835	2.029	1.162	4,671	.99	32,500	7.490	15.55	48.2
	196.4	4,929	.700	.404	1.629	.933	1.624	.930	2.329	1.337	4,932	1.00	34,360	8.750	17.65	49.5
	216.0	5,421	.716	.413	1.991	1.140	1.947	1.120	2.707	1.553	5,400	.99	37,580	11.70	23.07	50.7
D .358 D. W. 40¼ in. diam. 8.85 sq. ft.	47.8	1,199	.040	.023	.102	.058	.102	.058	.142	.081	1,246	1.04	11,030	.174	.45	38.8
	72.5	1,819	.070	.040	.194	.111	.192	.110	.264	.151	1,695	.93	15,010	.455	1.30	35.0
	99.0	2,459	.080	.046	.340	.196	.341	.196	.420	.242	2,260	.92	20,010	1.070	2.80	38.2
	113.8	2,856	.410	.236	.479	.275	.417	.275	.889	.511	2,673	.94	23,670	1.775	4.24	41.8
	117.2	2,941	.290	.167	.458	.264	.457	.263	.748	.431	2,616	.89	23,260	1.675	4.25	39.4
	131.6	3,223	.400	.231	.613	.352	.626	.360	1.013	.583	3,004	.93	26,600	2.555	5.19	49.2
	140.4	3,524	.500	.288	.746	.428	.740	.426	1.246	.716	3,329	.94	29,470	3.445	8.19	42.1
	151.1	3,732	.485	.279	.746	.428	.749	.430	1.231	.707	3,349	.90	29,650	3.460	8.91	38.8
	167.6	4,206	.500	.288	.875	.505	.880	.507	1.375	.793	3,630	.86	32,140	4.430	10.82	40.9
	189.6	4,758	....	....	1.160	.670	1.147	.660	....	....	4,144	.87	36,690	6.700	....	....
	191.4	4,804	.710	.409	1.259	.720	1.253	.720	1.969	1.129	4,332	.90	38,360	7.535	18.02	41.7
E .512 D. W. 48¼ in. diam. 12.65 sq. ft.	69.4	1,731	.090	.051	.136	.078	.133	.076	.226	.129	1.411	.81	17,850	.3795	1.31	28.9
	106.9	2,682	.230	.132	.266	.152	.260	.149	.496	.284	1,973	.73	24,960	1.035	3.76	27.5
	135.6	3,403	.374	.215	.413	.238	.403	.233	.787	.453	2,466	.72	31,200	2.025	7.14	28.4
	152.7	3,834	.456	.263	.480	.281	.486	.280	.945	.544	2,708	.71	34,260	2.630	9.34	28.2
	175.6	4,407	.630	.364	.644	.372	.638	.367	1.274	.736	3,102	.70	39,250	3.980	14.80	26.9
	210.0	5,271	.860	.495	.902	.520	.893	.513	1.762	1.015	3,657	.69	46,280	6.560	24.93	26.3
F .645 D. W. 54 in. diam. 15.90 sq. ft.	80.7	2,025	.120	.069	.136	.078	.133	.076	.256	.147	1.411	.70	22,420	.478	1.82	26.2
	117.1	2,938	.365	.211	.252	.144	.241	.138	.617	.355	1,900	.65	30,180	1.185	5.20	22.7
	122.7	3,079	.442	.255	.293	.168	.281	.162	.735	.423	2,051	.67	32,590	1.495	5.43	27.5
	164.1	4,118	.790	.456	.479	.287	.497	.273	1.269	.743	2,673	.65	42,490	3.330	14.10	23.6
	194.7	4,886	.944	.543	.638	.367	.597	.343	1.582	.910	2,990	.61	47,520	4.750	19.52	24.3
	202.4	5,080	1.130	.652	.772	.477	.719	.414	1.902	1.129	3,441	.68	54,680	7.110	24.48	29.0
G 0-D. W. outlet closed.	51.3	1,299	....	....	.200	.115	....	....	....	....	....	1.28	....	....	.34	....
	72.4	1,819	....	....	.297	.171	....	....	....	....	....	1.11	....	....	.59	....
	98.8	2,483	....	....	.477	.276	....	....	....	....	....	1.04	....	....	1.08	....
	129.2	3,247	....	....	.557	.318	....	....	....	....	....	.85	....	....	1.95	....
	140.0	3,518	....	....	.936	.540	....	....	....	....	....	1.02	....	....	2.21	....
	151.7	3,812	....	....	1.097	.633	....	....	....	....	....	1.02	....	....	2.77	....
	176.0	4,423	....	....	1.571	.907	....	....	....	....	....	1.06	....	....	4.33	....
	203.3	5,109	....	....	2.051	1.184	....	....	....	....	....	1.04	....	....	6.08	....
	226.8	5,699	....	....	2.481	1.433	....	....	....	....	....	1.03	....	....	8.11	....

DISCUSSION.

*Mr. E. S. Farwell.**—Since the preparation of this paper, the question has been raised of the relation said to exist between the

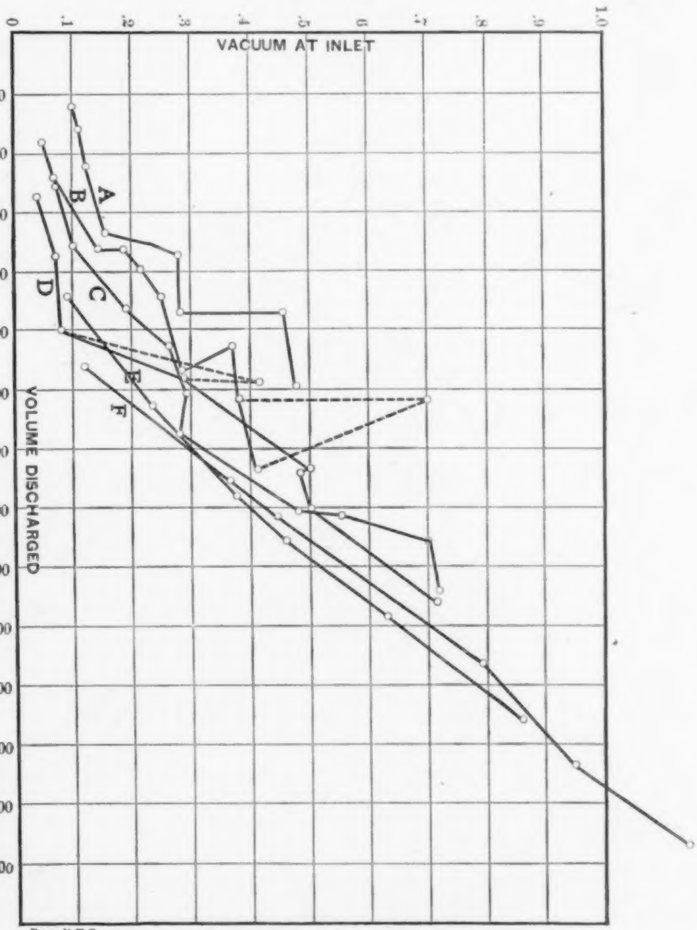


FIG. 40.

vacuum at inlet and the cubic feet of air discharged per minute. If the data were correctly taken, there should be a fixed relation between these two quantities. That relation is illustrated

* Added after adjournment.

in Fig. 40. Aside from a few unquestionably erratic points which have been connected to the proper curves by dotted lines, I think the curves show a very satisfactory relation existing between vacuum at inlet and cubic feet of air discharged.

The curves for tests E and F are certainly very interesting, and it may be pointed out that in these tests the vacuums at inlet were larger and the probable errors in reading of water gauge were proportionately less. The point which I attempted to bring out in paragraph 6, viz., the limitations imposed by having a single inlet, I do not think could have been presented so clearly, if vacuum at inlet had not been read.

No. 1019.*

A SERIES DISTILLING APPARATUS OF HIGH EFFICIENCY.

BY W. F. M. GOSS, LAFAYETTE, INDIANA.
(Member of the Society.)

1. This apparatus is designed to purify water or other liquids by distillation. It can be used, also, in the concentration of liquids carrying solids in solution, the recovery of which is desired. It consists of an arangement whereby the supply of liquid to be evaporated passes through a succession of chambers, in each of which it is gradually raised in temperature, and in all but the first of which a portion of the supply is vaporized, the process continuing until all has been changed to vapor. Heat is supplied the system at the chamber having the highest temperature only, from which vapor starts in a return circulation, that generated in one element of the apparatus serving as the source of heat for the element next lower in temperature, the liquid steam increasing in volume as it passes the successive elements until, finally, all is discharged from the chamber of lowest temperature. The liquid thus discharged is entirely the result of condensation. An apparatus embodying this conception, designed to distil 500 gallons of water an hour with an efficiency of approximately 60 pounds of water per pound of coal, is shown by Fig. 41, and a section of a single element entering into its construction, by Fig. 42.

2. The action of the apparatus may be best described in connection with the diagrammatic sketch Fig. 43. Several chambers, *A*, *B*, *C* and *D*, each containing a central tube, are connected by suitable piping. The first chamber, *A*, is entirely filled with liquid and constitutes a preliminary heater. The remaining chambers, from *B* to *D* inclusive, are partially filled with liquid, and from each of these, evaporation occurs. The heater and the cham-

* Presented at the New York meeting, December 1903, of the American Society of Mechanical Engineers, and forming part of Volume XXV. of the *Transactions*.

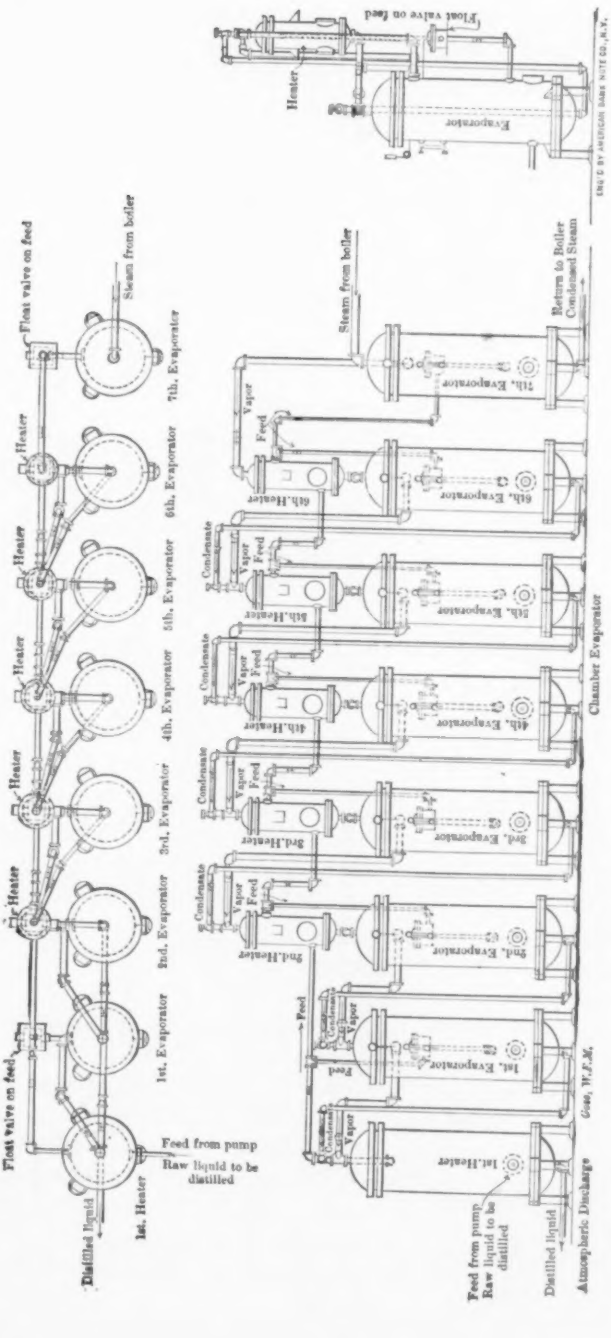

Float valve on feed
Heater
Evaporator
Float valve on feed
Steam from boiler
7th. Evaporator
6th. Evaporator
5th. Evaporator
4th. Evaporator
3rd. Evaporator
2nd. Evaporator
1st. Evaporator
Heater
Feed from pump
Raw liquid to be distilled
1st. Heater
Distilled liquid
Steam from boiler
Return to Boiler
Condensed Steam
Vapor
Feed
Condensate
7th. Evaporator
6th. Heater
6th. Evaporator
5th. Heater
5th. Evaporator
4th. Heater
4th. Evaporator
3rd. Heater
3rd. Evaporator
2nd. Heater
2nd. Evaporator
1st. Evaporator
1st. Heater
Chamber Evaporator
Feed from pump
Raw liquid to be distilled
Distilled liquid
Atmospheric Discharge
ENG'D BY AMERICAN BANK NOTE CO., N.Y.

FIG. 41.

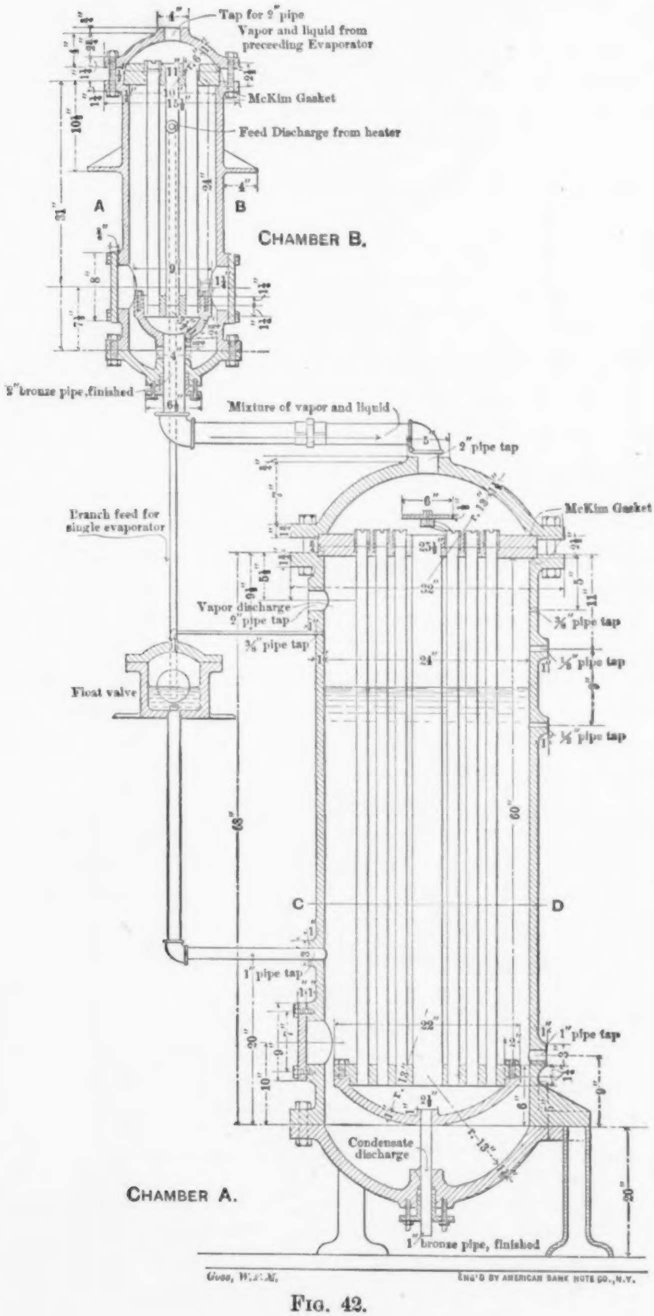

Tap for 2" pipe
Vapor and liquid from preceeding Evaporator
McKim Gasket
Feed Discharge from heater
A
B
CHAMBER B.
2" bronze pipe, finished
Mixture of vapor and liquid
2" pipe tap
McKim Gasket
Branch feed for single evaporator
Vapor discharge 2" pipe tap
¾" pipe tap
¾" pipe tap
½" pipe tap
½" pipe tap
Float valve
C
D
1" pipe tap
1" pipe tap
Condensate discharge
1" bronze pipe, finished
CHAMBER A.

FIG. 42.

ber B operate under one pressure, and each succeeding chamber, as C and D, under higher pressures. Heat is supplied to the liquid in each chamber by vapor or a mixture of vapor and liquid in the central tube, the external source of heat being H, which may be considered a small boiler having a closed circulation, steam being delivered by a vertical pipe, G, condensed in the central tube of the chamber D and returned to the boiler H.

3. The water to be evaporated is delivered into the bottom of the chamber A, and is gradually heated by contact with the central tube, as it arises to the top, from which point it passes to the

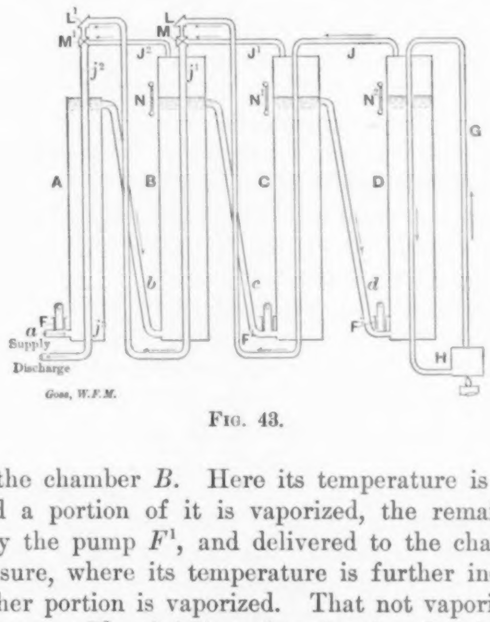


FIG. 43.

bottom of the chamber B. Here its temperature is further increased and a portion of it is vaporized, the remainder being taken up by the pump F^1, and delivered to the chamber C, of higher pressure, where its temperature is further increased and where another portion is vaporized. That not vaporized in C is taken up by pump F^2 and delivered to chamber D, which is under a still higher pressure, where it is entirely vaporized. The return circulation begins with the vapor given off in the chamber D, which passes off by the pipe J and becomes a source of heat for the chamber C, being condensed by the time it reaches the bottom of chamber C. From this point it is conveyed to the reducing pressure valve L, where its pressure is reduced to the pressure of the vapor within the chamber C. From L the stream passes on to the mixing valve M, where it intermingles with the steam delivered from the chamber C, the combined stream of liquid and vapor passing through the central tube of the chamber B, serving

as its source of heat, being condensed to liquid form and, finally, being delivered to the reducing valve L^1, where its pressure is again reduced, and from which point it passes to M^1, where it mingles with the vapor given off by the chamber B, after which the combined streams of vapor and liquid pass through the central tube of chamber A, are cooled and finally discharged.

4. In the process thus briefly outlined, the following facts are to be noted:

a. That the ingoing stream of liquid is continually increasing in temperature, and that the return current of vapor is continually diminishing in temperature, the arrangement being such that it is entirely possible to have a fixed difference between the heating medium and the medium to which heat is imparted throughout all portions of the apparatus; also, each particular portion of each element retains a fixed temperature.

b. The losses of heat which occur from the apparatus are those of radiation and those which are represented by the differences in the temperature or condition between the liquid supplied and the condensate delivered. Heat thus lost must be supplied from the source of heat and nothing more.

c. It is conceivable that the temperature of the discharge may be so reduced as to approach very nearly the temperature of the supply; also, that as ideal conditions are approached, radiation losses may be diminished, and, at the limit, losses on both of these accounts cease; that is, the cycle of the apparatus is one of maximum efficiency.

5. For satisfactory operation, the total range of temperature to which the apparatus is subjected is divided equally between the several stages of the process, the temperature of each chamber being controlled by the pressure which is maintained upon it, which in turn is regulated by valves on outlets. For example, if the apparatus shown diagrammatically by Fig. 43 is assumed to be operated from a temperature of 360 degrees in the boiler H, to a final temperature in chamber B of 212, then the temperature of the water in chamber C would be maintained at 262, and that of the water in chamber D at 312, the difference in temperature for each transmission being in this case 50 degrees. This value also measures the difference in temperature between the mixture within the central tube and the liquid around the same in each chamber. The pressures upon the several chambers, etc., assuming the liquid operated upon to be water, would be, for the boiler H,

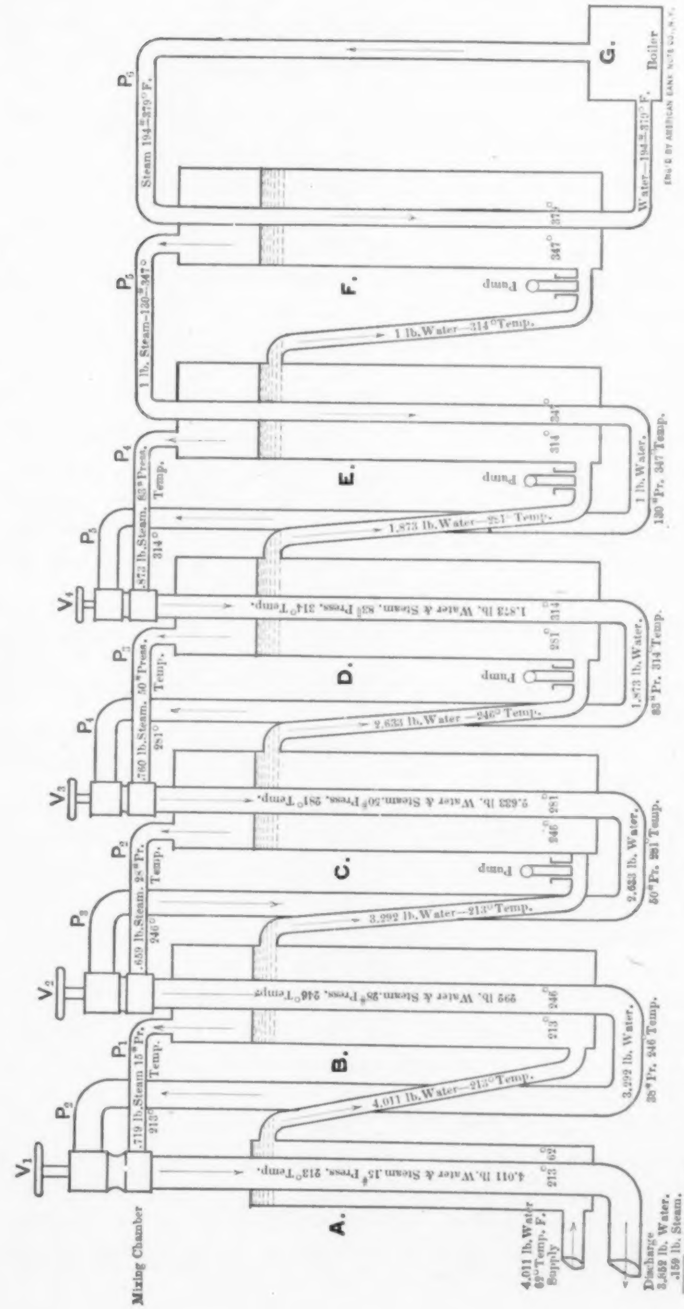

A.
B.
C.
D.
E.
F.
G.
Boiler
Mixing Chamber
Pump
4,011 lb. Water 62° Temp. F. Supply
Discharge 3,852 lb. Water. .159 lb. Steam. 4,011 Water & Steam. 15# Press. 213° Temp.
4,011 lb. Water & Steam 15# Press. 213° Temp.
4,011 lb. Water—213° Temp.
3,292 lb. Water—213° Temp.
2,633 lb. Water—246° Temp.
1,873 lb. Water—281° Temp.
1 lb. Water—314° Temp.
Goss, W.F.M.

FIG. 44.

145 pounds; the chamber *D* and pipe *J*, 65 pounds; the chamber *C* and pipe J^1, 25 pounds; the chamber *B* and pipe *J* , atmosphere; the pressure in each case being controlled by valves upon the discharge pipe.

6. The action thus far described in general terms is susceptible of careful analysis, some of the results of which appear in Fig. 44. This figure is a diagram of a five-stage apparatus, upon which are noted the pressure, temperature and condition of mixture of the circulating streams in their course through the system; also, the weight of mixture transmitted from one chamber to another for each pound of vapor delivered by the chamber of highest temperature, *F*. Other assumptions underlying the values given are as follows:

a. That the liquid distilled is water.

b. That the difference of temperature between each step in the process is 33 degrees; that is, that the temperature of each succeeding chamber is 33 degrees higher than that of the chamber next below, from which it follows that in any given element the temperature within the central tube is 33 degrees higher than the temperature of the surrounding liquid.

c. That the temperature of water supplied is 62 degrees.

d. That the discharge from *A* is at atmospheric pressure; that is, at a pressure of 15 pounds, absolute, and a temperature of 213 degrees.

e. That 2 per cent. of the heat supplied each chamber is lost by radiation.

7. A more detailed presentation of analytical results is reserved for later paragraphs, but it may here be noted that the efficiency of the apparatus depends upon the number of chambers employed, and that its characteristic feature is to be found in the fact that it so conserves the heat supplied it that the process may be successfully extended through a long series of steps.

8. Considering now the details of the apparatus, it is to be observed that the process thus far described involves a separate pump for each of the several evaporator chambers. Thus, in Fig. 43, there are provided for the evaporator chambers *B*, *C* and *D* the pumps *F*, F^1 and F^2. This necessity arises from the fact that at each stage the feed is passing from a lower to a higher pressure, and the requirements of the cycle necessitate that the water supplied to each pump shall have the temperature of the chamber next below before being forced to the

succeeding one. While this requirement is not serious when the number of evaporator chambers is small, it leads to objectionable complication when the series is extended. In the practical working out of the design, therefore, a system of feeding is employed which allows the use of a single pump, without impairing the thermodynamic cycle, the modified arrangement being inferior to that already described only in the fact that it adds slightly to the mechanical work necessary to move the feed. Its single feed-pump delivers water at a pressure sufficiently high to supply all evaporator chambers, including that of highest pressure. Water delivered by the pump is passed through a succession of heaters connected with the evaporator chambers in such a manner that the feed emerging from each heater has the same temperature as it would have had if it mingled with the water of the chamber, while branch pipes are employed to convey the feed to each individual chamber. The course of the piping in its external appearance is shown by Fig. 41. It will be seen that the feed enters the base of the preliminary heater, is delivered therefrom at a point near the top, is conveyed to a small heater over the second evaporator chamber, thence to the heater over the third evaporator chamber, and thence on to the last heater over the sixth evaporator chamber. Branches are taken off after each heater to supply the evaporator chamber with which the heater is connected. The control of this supply is best indicated by Fig. 42, showing the float-valve regulating the admission of water to the evaporator chamber. It will be remembered that the pressure above the float-valve is greater than the pressure in any of the evaporator chambers, so that the feed enters whenever the valve is off its seat. Referring more particularly to the heater (Fig. 42), it will be seen that this is similar in construction with the main evaporator chamber. The mixture of vapor and liquid which is to supply heat to the main evaporator chamber first passes through the tubes of the heater, the feed water circulating around the tubes.

9. Concerning the maintenance of the apparatus, it should be observed that the boiler from which the heat is supplied (Fig 43, *H*) is subject to a closed circulation. No renewal of water is required, and hence there can be no trouble at this point from incrustation. Concerning deposits of solid matter in other portions of the system, it is to be noted that only distilled water and vapor circulate within the tubes (Fig. 42). This is true of the heater

as well as of the evaporator chamber. The solid matter which may be deposited within the chamber around the tubes may be disposed of by use of a blow-off, by cleaning through manholes, or by the complete removal of the nests of tubes from the shell of the chamber. In the construction of the apparatus the central tube is threaded to the tube-plate at both ends. All other tubes are threaded at their lower ends only. The presence of the central tube maintains the relative position of the tube-plates and permits all the tubes of a chamber to be withdrawn together.

10. Having defined the character of the apparatus, attention may be given to matters affecting its performance. In general, three different classes of the apparatus are to be dealt with, and for convenience Class A will be designated as representing an apparatus all portions of which are operated at pressures above the atmosphere; Class B as representing an apparatus all portions of which are to operate below the pressure of the atmosphere, and Class C as representing an apparatus a portion of which is to operate at pressures above the atmosphere and a portion at pressures below the atmosphere. Class A may be regarded as a high-pressure apparatus, Class B a low-pressure apparatus such as might be operated by the use of exhaust steam, and Class C a high and low pressure apparatus. Practical and convenient ranges of pressure and temperature applying to each of the three classes thus designated, when designed for the distillation of water, are as follows:

TABLE I.

Designation.		Absolute Pressure.		Temperature F.	
Class.	As to Pressure.	Highest.	Lowest.	Highest.	Lowest.
A	High pressure.	115	15	338	212
B	Low pressure.	15	1.0	212	102
C	High and low pressure.	115	1.0	338	102

The relations shown are merely chosen for purposes of illustration. So far as the working of the apparatus is concerned, any other practical limits might have been chosen.

11. The weight of water which will be distilled for each pound of saturated steam supplied from the source of heat for an apparatus of the high-pressure type, working under the conditions of

pressure set forth in Table I., and assuming no loss to occur by radiation or in the form of vapor from the delivered stream, will be as follows:

TABLE II.

WEIGHT OF DISTILLED WATER PER POUND OF STEAM USED.

Number of Evaporator Chambers.	Pounds of Water per Pound of Steam.
3	2.8
4	3.6
5	4.4
6	5.2
7	5.9
8	6.5
9	7.1
10	7.6

12. The weight of water distilled for each pound of steam supplied is nearly the same for all three classes as above defined. An approximate relation applying to all classes is represented by the formula, $W = 0.85\ N$, in which W is the number of pounds of water distilled for each pound of steam supplied, and N is the number of evaporator chambers. If the boiler which serves as the source of heat delivers ten pounds of steam per pound of coal burned, the output of distilled water per pound of coal will be ten times the values given in Table II.

13. Results appearing in the preceding Table II. are in some cases subject to correction to cover losses due to the presence of vapor in the discharged stream. In the action of the machine, all condensation is the result of the cooling action of the incoming stream of liquid feed. Under certain conditions, this will be insufficient, in which case a portion of the issuing stream delivered from the apparatus will be vapor and will in part disappear as it emerges from the pipe. The extent of this loss, if any, decreases as the number of chambers is increased. In apparatus of the Class A it entirely disappears when seven evaporator chambers are used; but in types B and C there will be a slight loss even when as many as ten evaporator chambers are used. The percentage of the total weight of the discharged stream which will be delivered in the form of vapor is as follows:

TABLE III.

PERCENTAGE OF DISCHARGED STREAM LOST BY VAPORIZATION.

Number of Evaporator Chambers.	Class of Apparatus.		
	A	B	C
3	21	33	32
4	11	25	22
5	6	19	16
6	2	16	12
7	0	13	10
8	0	11	8
9	0	9	7
10	0	8	6

14. The extent of heating surface in each element (one evaporating chamber and its attached heater) needed for a given capacity is not materially affected by changes in the number of elements in the series. Assuming the pressure ranges set forth in Table I., and assuming each foot of tube surface to transmit 400 thermal units per degree difference of temperature per hour, there will be required in each element of an apparatus designed to distil 1,000 gallons of water per hour an amount of heating surface as follows:

TABLE IV.

SQUARE FEET OF TUBE SURFACE IN EACH ELEMENT PER 1,000 GALLONS OF WATER TO BE DISTILLED.

Class of Apparatus.	Tube Surface.
A (High Pressure).	174
B (Low Pressure).	188
C (High and Low).	113

15. The total amount of tube surface required for any apparatus of the capacity stated will be found by multiplying the number of evaporator chambers it is to contain, plus one, by the value assigned its class in the preceding statement. The unit added covers the large heater (1st Heater, Fig. 41). For example, if it is required to design a seven-chamber apparatus of the A class to distil 1,000 gallons of water per hour, then the total number of square feet of heating surface for the whole system will be 8 × 174 or 1,392 square feet.

16. The size of the small feed-water heaters varies with the

quantity of water which each must handle, and this diminishes with each step that the feed is advanced, the percentage of the total feed handled by each heater being as follows:

TABLE V.

PERCENTAGE OF THE TOTAL FEED PASSING EACH HEATER.

(Numbers assigned heaters agree with those given in Fig. 1.)

Number of Evaporator Chambers in Series.	Designation of Heater. (Number of heaters one less than number of evaporators.)								
	1	2	3	4	5	6	7	8	9
1	100								
2	100								
3	100	71							
4	100	76	53						
5	100	81	63	44					
6	100	84	70	54	37				
7	100	85	75	62	48	33			
8	100	86	75	64	52	40	27		
9	100	88	78	68	58	48	37	25	
10	100	90	81	72	63	53	43	33	23

17. From values in the preceding table, and the known capacity of a proposed apparatus, the weight of water in pounds which will pass each heater can be ascertained, after which the tube surface of each small heater may be found by use of the following formula, $A = 0.00173\, W$, in which A is the required area of tube-surface in the heater and W is the number of pounds of water passing the heater per hour. On the basis of this formula, it can be shown that the area of tube surface for the several small heaters of a seven-stage apparatus, designed to distil 1,000 gallons of water per hour, under the pressure range of Class A, Table I., should be as follows:

TABLE VI.

SQUARE FEET OF TUBE SURFACE IN SMALL HEATER.

Designation of Heater.	Square Feet of Heating Surface.
2d	4.8
3d	6.8
4th	8.8
5th	10.6
6th	12.4

This will serve to indicate the significance of the small heater as a part of the general design. Since all are relatively small, the action of the apparatus as a whole can not be greatly affected if all are made the same size. If in the preceding example all were made equal to the mean value obtained by calculation, the surface of each would be 8.6 square feet, and the area assigned the several

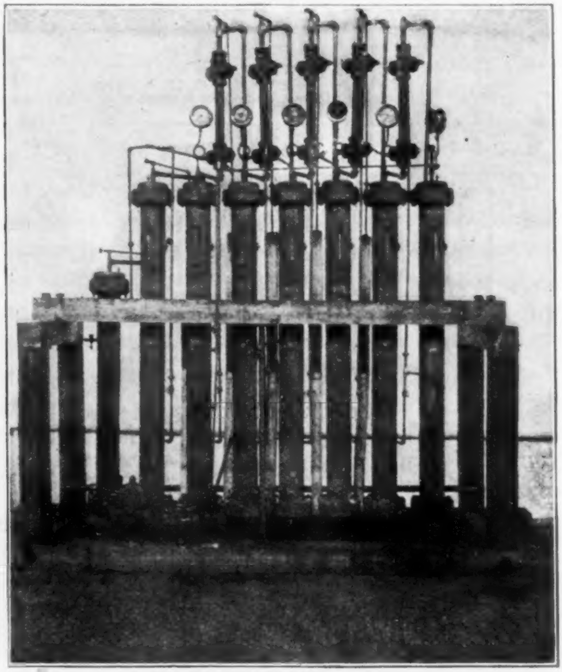


FIG. 45.

evaporator chambers (Table IV.) may be reduced by a like amount.

18. Having now determined the proportions which must be given the several members of the apparatus, it should be stated that the arrangement can not always be that set forth in Figs. 41 and 42. In a low-pressure system, or in a high-and-low pressure system (classes B and C), some modification in the mechanical arrangement shown by the diagrams is necessary. Thus, the chambers which are low in the scale of pressures must be so arranged that the water of condensation from one chamber will flow freely

to the chamber next lower in pressure. That is to say, it will be necessary to place one element above the other. The reason for such an arrangement is that the pressure difference between each chamber is not sufficient to overcome the head of water equivalent to the distance from the bottom of one chamber to the top of the next. Just how many chambers will need to be so arranged can only be stated when the conditions of the individual case shall have been defined. Obviously, the cycle will not be interfered with or the working of the apparatus changed if all the elements are arranged in a vertical series instead of a horizontal series.

19. To the foregoing should be added the fact that the practicability of the cycle described has been well established by the performance of a small seven-chamber apparatus, which is shown by Fig. 45. The view makes the machine appear more complicated than it really is, since the heaters are made up of pipe fittings. It shows also the apparatus with most of the covering removed, whereas in service all portions of the machine were covered with non-conducting material. By the aid of this apparatus it has been shown that vaporization takes place and the whole action proceeds as has been described, even when the temperature difference between adjacent chambers is less than ten degrees, and by its use facts already presented, with reference to capacity and efficiency, have been substantially checked.

In conclusion, the writer finds pleasure in acknowledging the important assistance rendered both in the analytical study and in the experimentation which followed it, by Prof. R. S. Miller and Mr. Charles Ducas, Junior Members of the Society.

DISCUSSION.

Mr. H. H. Suplee.—I should like to ask wherein this arrangement of evaporators differs from the multiple-effect evaporating pan system of Rillieux, as used in the manufacture of sugar. I believe that some of the vessels of the United States navy are equipped with multiple-effect evaporators which are practically the same as those used in sugar works, and, so far as I can recall, the arrangement is practically the same as that described in the paper.

Prof. D. S. Jacobus.—It is stated that the losses of heat which occur are those of radiation and those represented by the differences in the temperature or condition between the liquid supplied

and the condensate delivered. In testing a large Yaryan triple-effect evaporator, I found that, in order to get rid of air which would otherwise accumulate, we had to lead some steam from each effect directly into the one below it by partly opening by-pass valves provided for the purpose. If the air were allowed to accumulate, it would reduce the capacity of the evaporator. This naturally lowered the efficiency. Will Professor Goss kindly tell us whether he has noticed any similar action in the operation of his apparatus?

Prof. W. F. M. Goss.—Responding to the question of Mr. Suplee, I would say that in all other multiple-effect evaporating systems with which I am familiar the condensation from each pan or chamber is drawn out separately, cooled and delivered. The presence of the cooler leads to large losses of heat, and the series can not be made to include very many effects. In the apparatus under consideration, the entire discharge, not of vapor only, but of condensate as well, of one chamber is carried on to heat the chamber of the next lower temperature, and at the end of the series the distillate of all chambers is delivered and cooled in the form of a single stream.

Our experiments have not developed any difficulty of the kind referred to by Professor Jacobus. The apparatus illustrated in Fig. 45 has been kept in continuous operation for periods of twelve or fourteen hours, with no diminution in capacity or efficiency.

Professor Jacobus.—If the apparatus were operated with a vacuum at the last effect, the difficulty with the air might have been experienced. In the tests of the Yaryan evaporator there was a high vacuum at the last effect, and the capacity was considerably reduced if it was operated an hour or so without opening the by-pass valves.

*Professor Goss.**—It is true, as has been suggested, that all of our experiments have been with a high-pressure system. We have not used a vacuum. A study of the course of the feed and of the distillate through this apparatus is, however, reassuring, since the distillate from all sources is constantly leading on in a single direction, and will, I think, carry with it a reasonable amount of entrained air. For this reason, I do not anticipate trouble of the sort described, at whatever pressure the apparatus may be used.

It has been suggested that the value of the paper would be in-

---

* **Author's Closure under the Rules.**

creased if there were added a statement concerning the cost of distilling water by means of the apparatus described. The cost necessarily depends somewhat upon local conditions. For each dollar cost per ton of coal, a seven-chambered apparatus of the high-pressure type requires 7.1 cents' worth of fuel per thousand gallons of water distilled. Assuming such an apparatus to work 300 days in the year, and to cost \$3 per foot of tube surface, the interest and depreciation charges at 10 per cent., the first cost amounts to something less than 6 cents per thousand gallons, making the total cost of distilled water, excluding attendance, which may be little or much, 13 cents per thousand gallons. A general statement of these facts is as follows:

Cost in cents per thousand gallons = 7.1 × cost of coal per ton in dollars + 6.

Since the apparatus requires no cooling water, no costs can arise on this account, a fact which, under some conditions of service gives the series apparatus an important advantage over others of more simple form.

The following tabulated statement constitutes a more elaborate presentation of the facts affecting cost as based upon an apparatus of a thousand gallons per hour capacity:

1. Number of evaporating chambers.	4	7	10
2. Gallons of water distilled per year, assuming 300 working days of 24 hours each.	7,200,000	7,200,000	7,200,000
3. Pounds of coal per year.......	1,670,000	1,019,000	790,000
4. Cost of coal at one dollar per ton, dollars..	855	510	395
5. Area of tube surface in plant, feet	770	1,392	1,914
6. Cost of complete plant on the basis of three dollars per foot of tube surface, dollars..	2,210	4,176	5,742
7. Annual interest and depreciation, 10 p. c..	221	417	57[illegible]
8. Annual total cost, excluding attendance...	1,076	927	970
9. Fuel cost per thousand gallons for each dollar cost per ton of coal, cents........	11.8	7.1	5.0
10. Interest and depreciation charges per thousand gallons, cents.......	3.1	5.7	7.9
11. Fuel, interest and depreciation per thousand gallons, coal at one dollar per ton, cents..................................	14.9	12.8	13.4
12. Fuel, interest and depreciation per thousand gallons, coal at two dollars per ton, cents...............................	26.7	19.9	18.9
13. Fuel, interest and depreciation per thousand gallons, coal at three dollars per ton, cents...........................	38.5	27.0	24.4

No. 1020.*

THE PRESSURE TEMPERATURE CURVE OF SULPHUROUS ANHYDRIDE (SO_2).

BY EDWARD F. MILLER, BOSTON, MASS.
(Member of the Society.)

1. The use of sulphurous anhydride as one of the working vapors in the "Binary Heat Engine" or "Waste Heat Engine" has made evident the need of more complete tables of the properties of this vapor, especially at the high pressures.

2. A series of articles by Prof. R. H. Thurston printed in the Journal of the Franklin Institute, 1902, explains fully the principle of this engine.

3. A paper by the writer, read before the New England Water Works Association, printed in the 1902 proceedings and reprinted in *Engineering News* of November 27, 1902, gives an account of the engine at the Technische Hochschule at Charlottenberg with results of tests made on the engine.

4. Volume XII. of *Transactions* of A. S. M. E. contains the results of an experimental determination of the latent heat of SO_2 by Professor Jacobus.

5. Messrs. Flowers and Walton, of Cornell, tabulated the work of the various experimenters on SO_2 and from this tabulation constructed tables of the properties of this vapor. These tables are the most complete of any the writer knows of.

6. The work presented in this paper was carried on under the direction of the writer by Mr. D. D. Mohler, a senior in the course in Chemical Engineering at Mass. Inst. Technology.

7. The apparatus in which the SO_2 was vaporized consisted of an aluminum bronze cylinder with hemispherical end and cover, 7½ inches inside diameter and 10 inches in length inside of cover.

---

* Presented at the New York meeting (December, 1903) of the American Society of Mechanical Engineers, and forming part of Volume XXV. of the *Transactions*.

This cylinder was placed on end inside of a copper kettle 15 inches in diameter and 15 inches tall, which was open at the top. The kettle was covered on the outside with magnesia to a depth of 2 inches, and the bottom rested on an electric stove.

8. Cylinder oil filled the space between the kettle and the cylinder to a depth of 5 inches.

9. Immersed in the oil there was a heating coil of 25 feet of No. 24 B. & S. gauge iron wire and a stirrer with vanes turned so as to drive the oil from the top downwards. This stirrer was driven by a $\frac{1}{10}$ horse-power motor.

10. The cover of the aluminum bronze cylinder was provided with a gauge connection, a filling valve and a thermometer well reaching down into the liquid SO

11. Temperatures of the SO_2 were determined by an Alvergnait millimeter thermometer graduated to $\frac{1}{10}$ of a degree Fahr.

12. Pressures were measured by a gauge of large diameter provided with a steel tube made specially for this purpose.

13. This gauge could be read to $\frac{1}{10}$ of a pound.

14. The SO_2 used was found by chemical analysis to be 99.3 per cent. pure with .3 per cent. of CO_2 and .4 per cent. air.

15. In filling the aluminum bronze cylinder with SO_2 the air was first exhausted to a vacuum of 28 inches. Liquid SO_2 was then run in till the pressure inside the cylinder was the same as that of the atmosphere. The cylinder was then exhausted again to 28 inches vacuum and more SO_2 supplied. The operation was repeated a third time.

16. After the air had been removed eight pounds of liquid SO_2 were run in. This amount brought the level of the liquid SO_2 a little above that of the cylinder oil in the outer kettle.

17. The oil could be heated sufficiently in from 10 to 15 minutes by the resistance coil to cause an increase of pressure of 5 pounds inside the cylinder.

18. After heating to the desired temperature the current was turned from the coil and the electric stove under the kettle was used to supply sufficient heat to make up for that lost by radiation.

19. By this means the pressure could be maintained nearly constant for a considerable space of time.

20. A series of preliminary observations gave sufficient data to enable one to plot the curve with considerable accuracy.

21. The final observations, covering a continuous period of 24 hours were taken as follows: At low temperature 20 to 25 readings

were taken at each pressure at 15 second intervals. The thermometer readings for each set were practically constant: when the higher pressures were reached, where the temperature change is slight for a considerable change of pressure, it was found that the thermometer lagged; the gauge showing a change before any movement of the mercury could be detected.

22. To allow for this lag of the thermometer readings were begun when the gauge had reached within $\frac{1}{10}$ of a pound of the maximum pressure and were continued at 15 second intervals till the pressure had fallen to $\frac{1}{10}$ pound below the maximum pressure.

23. It was assumed that the lag of the thermometer on a falling pressure and falling temperature would be the same as the lag on a rising pressure and rising temperature.

24. In some cases as many as 40 observations were taken on one determination.

25. The observations corrected for the lag of the thermometer and for errors in the thermometer and in the gauge are given below:

TABLE I.

SUMMARY OF CORRECTED OBSERVATIONS.

Temperature °F.	Pressure Lbs. Absolute.	Temperature °F.	Pressure Lbs. Absolute.	Temperature °F.	Pressure Lbs. Absolute.	Temperature °F.	Pressure Lbs. Absolute.
68.20	49.41	131.93	138.70	171.00	243.61	198.71	352.12
74.88	55.35	139.11	154.40	172.78	249.65	200.01	358.01
81.04	61.77	142.86	160.92	176.51	262.71	201.21	363.80
87.05	68.80	144.06	166.45	178.97	270.96	202.33	368.76
92.22	76.07	146.31	173.20	180.33	276.78	203.05	371.84
98.37	82.56	148.65	178.46	183.32	287.50	204.97	381.18
103.12	89.59	151.13	184.59	185.01	294.73	206.49	388.04
106.20	94.72	154.37	193.12	187.51	303.26	207.24	392.48
109.62	99.77	156.63	200.01	189.24	311.26	208.26	398.08
112.32	104.57	158.29	204.80	191.29	317.21	209.80	405.89
115.05	109.01	161.94	215.58	192.58	323.64	210.41	409.72
124.72	125.75	164.22	222.73	194.19	330.54	211.14	416.74
128.74	133.29	166.71	229.21	195.23	337.25		
129.95	137.17	168.53	236.03	197.14	344.70		

26. The writer has plotted both these observations and those representing Regnault's curve on plotting paper 11 feet by 4 feet

6 inches and after drawing a smooth curve through the points has read from this curve the pressure corresponding to each degree.

27. At high pressure the work may be in error possibly one pound, at lower pressure the error is much less.

28. The following table will enable one to compare the results obtained by the various experimenters:

TABLE II.

Temperature °F.	Pressures from the Tabulation of Messrs. Flowers and Walton.	Pressures. Experiments of Sajotschewski.	Pressures. Experiments of Blumcke.	Pressures. Experiments of Regnault.	Pressures from Experiments given in this Paper.	Temperature °F.	Pressures from the Tabulation of Messrs. Flowers and Walton.	Pressures. Experiments of Sajotschewski.	Pressures. Experiments of Blumcke.	Pressures. Experiments of Regnault.	Pressures from Experiments given in this Paper.
68	47.7	.....	.....	47.6	47.9	149	177.8	.....	180.6	178.0	179.5
77	56.4	.....	.....	56.4	56 6	158	200.8	210.4	.....	.....	203.8
86	66.4	.....	.....	66.4	66.7	167	226.1	.....	.....	.....	230.7
95	77.7	.....	80.1	77.6	78.0	171.5	.....	....	251.7	.....	245.2
104	90.3	.....	.....	90.4	90.6	176	253.5	265.9	.....	.....	260.5
113	104.4	.....	.....	104.5	104.7	194	.....	..330 3	.....	.....	331.1
116.1	.....	.....	111.0	.....	109.9	209.3	.....	....	396.3	.....	402.0
122	120.1	123.9	.....	120.4	120.5	212	.....	..408.9	.....	.....	418.0
131	137.5	.....	.....	137.9	137.9	248	....	610.9	.....	.....	.....
140	156.7	163.0	.....	157.1	157.6	302	.....	1,050.3	.....	.....	.....

29. Regnault experimented between the temperatures—40 degrees Fahr. and 149 degrees Fahr.; Sajotschewski between 122 degrees Fahr. and 302 degrees Fahr., determining 8 points. This curve is above the others up to about 200 degrees Fahr. where it apparently crosses; Blumcke between 95 degrees Fahr. and 209.3 degrees Fahr., determining 5 points.

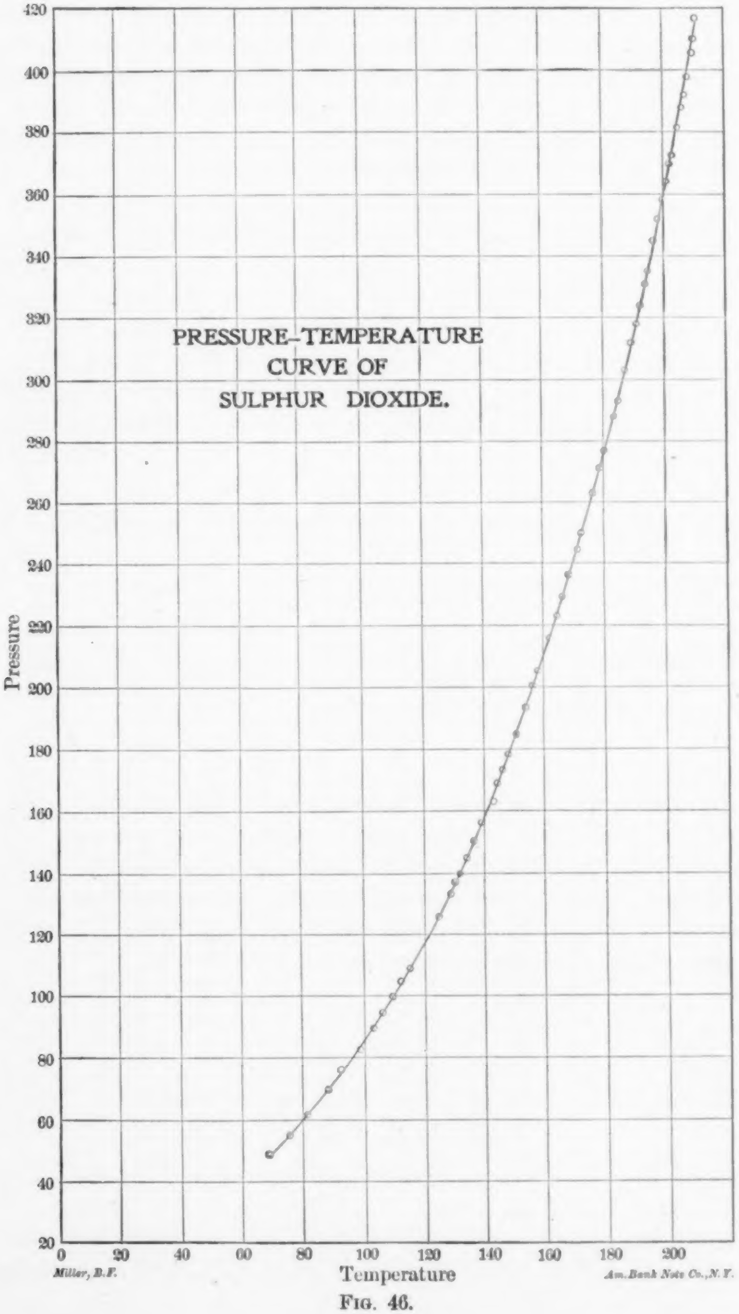

PRESSURE–TEMPERATURE
CURVE OF
SULPHUR DIOXIDE.
Pressure
420
400
380
360
340
320
300
280
260
240
220
200
180
160
140
120
100
80
60
40
20
0
20
40
60
80
100
120
140
160
180
200
Temperature
Miller, B.F.
Am. Bank Note Co., N.Y.

FIG. 46.

TABLE III.

Saturated Sulphurous Anhydride.

Temperatures and Pressures Corresponding as Read from the Plot; no Attempt Made to Correct Values by Making Use of a Table of Second Differences.

Temperature °F.	Pressure Lbs. Absolute.	Temperature °F.	Pressure Lbs. Absolute.	Temperature °F.	Pressure Lbs. Absolute.	Temperature °F.	Pressure Lbs. Absolute.	Temperature °F.	Pressure Lbs. Absolute.	Temperature °F.	Pressure Lbs. Absolute.
-40	3.1	3	11.2	46	30.7	89	70.3	132	139.9	175	257.0
-39	3.2	4	11.5	47	31.4	90	71.6	133	141.9	176	260.5
-38	3.3	5	11.8	48	32.1	91	72.8	134	143.9	177	264.0
-37	3.4	6	12.1	49	32.8	92	74.1	135	146.0	178	267.6
-36	3.6	7	12.4	50	33.5	93	75.4	136	148.2	179	271.2
-35	3.7	8	12.7	51	34.2	94	76.7	137	150.5	180	274.9
-34	3.8	9	13.1	52	34.9	95	78.0	138	152.8	181	278.6
-33	4.0	10	13.4	53	35.6	96	79.3	139	155.2	182	282.4
-32	4.1	11	13.7	54	36.4	97	80.7	140	157.6	183	286.3
-31	4.2	12	14.0	55	37.1	98	82.1	141	159.9	184	290.2
-30	4.3	13	14.4	56	37.9	99	83.5	142	162.3	185	294.2
-29	4.5	14	14.8	57	38.7	100	84.9	143	164.7	186	298.2
-28	4.7	15	15.2	58	39.4	101	86.3	144	167.1	187	302.2
-27	4.9	16	15.6	59	40.2	102	87.7	145	169.5	188	306.2
-26	5.0	17	16.0	60	41.0	103	89.1	146	171.9	189	310.2
-25	5.1	18	16.4	61	41.8	104	90.6	147	174.4	190	314.3
-24	5.2	19	16.8	62	42.6	105	92.1	148	176.9	191	318.4
-23	5.4	20	17.2	63	43.4	106	93.6	149	179.5	192	322.6
-22	5.5	21	17.6	64	44.3	107	95.1	150	182.1	193	326.8
-21	5.7	22	18.0	65	45.2	108	96.7	151	184.8	194	331.1
-20	5.9	23	18.4	66	46.1	109	98.3	152	187.4	195	335.4
-19	6.0	24	18.8	67	47.0	110	99.9	153	190.0	196	339.8
-18	6.1	25	19.3	68	47.9	111	101.5	154	192.7	197	344.2
-17	6.3	26	19.7	69	48.8	112	103.1	155	195.4	198	348.7
-16	6.5	27	20.1	70	49.7	113	104.7	156	198.2	199	353.2
-15	6.7	28	20.6	71	50.6	114	106.3	157	201.0	200	357.7
-14	6.9	29	21.0	72	51.6	115	108.0	158	203.8	201	362.3
-13	7.1	30	21.5	73	52.6	116	109.7	159	206.6	202	367.0
-12	7.3	31	22.0	74	53.6	117	111.4	160	209.5	203	371.8
-11	7.5	32	22.5	75	54.6	118	113.2	161	212.4	204	376.7
-10	7.8	33	23.1	76	55.6	119	115.0	162	215.3	205	381.6
-9	8.0	34	23.6	77	56.6	120	116.8	163	218.3	206	386.6
-8	8.3	35	24.2	78	57.7	121	118.6	164	221.4	207	391.7
-7	8.5	36	24.7	79	58.8	122	120.5	165	224.5	208	396.7
-6	8.7	37	25.3	80	59.9	123	122.4	166	227.6	209	401.8
-5	9.0	38	25.9	81	61.0	124	124.3	167	230.7	210	407.0
-4	9.3	39	26.5	82	62.1	125	126.2	168	233.9	211	412.5
-3	9.5	40	27.1	83	63.2	126	128.1	169	237.1	212	418.0
-2	9.7	41	27.7	84	64.3	127	130.0	170	240.3		
-1	10.0	42	28.2	85	65.5	128	131.9	171	243.6		
0	10.3	43	28.8	86	66.7	129	133.9	171	246.9		
1	10.6	44	29.4	87	67.9	130	135.9	173	250.2		
2	10.9	45	30.1	88	69.1	131	137.9	174	253.6		

DISCUSSION.

Mr. S. A. Moss.—Investigation of Professor Miller's table of final results, Table III., seems to indicate that he has given more weight to Regnault's values than to his own in the region where the experiments overlap. In this region Professor Miller's pressures, as shown by the unsmoothed values given in Table I., are higher than Regnault's. Would it not be better to give a table of the smoothed values actually obtained by Professor Miller instead of Table III., which gives values from 68 degrees to 149 degrees lower than he obtained?

It has been interesting to compare Professor Miller's values with those which would have been predicted by Ramsay's and Young's general law for vapor pressures. This law, which is not as generally known as its importance deserves, gives a means of computing the vapor pressure corresponding to any temperature, if a few values of corresponding pressure and temperature are known.

In a modified form (see *Physical Review*, Vol. XVI., No. 6, June, 1903) the law is as follows: Let Tx be the absolute temperature corresponding to any vapor pressure for substance x, and Tw the absolute temperature corresponding to the same pressure for water vapor, as given by Steam Tables. Then $\frac{1}{Tx} = a\frac{1}{Tw} - b$ where a and b are constants, different for each substance. The values given by Professor Miller in Table I. follow the law exactly up to about 157 degrees. The constants as computed for this region from Table I. give as the equation for SO_2,

$$\frac{1}{Ts} = 1.6667\frac{1}{Tw} - .00036.$$

Absolute zero is taken as $-$ 459.5 degrees Fahr. If Ramsay's and Young's law holds true for all values, this equation gives the saturation temperature for any pressure whatever if we know the saturation temperature for water vapor (steam) at that pressure.

As stated, the values given by Professor Miller in Table I. follow this law exactly up to 157 degrees. Beyond this point there is a slight departure, the pressures given by the law being lower than those given by Professor Miller. This departure is noth-

ing at 157 degrees and gradually increases. For 407 pounds pressure, Professor Miller gives 210 degrees, while the temperature predicted by the law is 215.2. This discrepancy indicates either that Ramsay's and Young's law does not hold for the higher pressures of SO_2 or else that Professor Miller's higher pressures are somewhat too great.

*Professor Miller.**—The values given by Regnault between 68 degrees and 149 degrees were given equal weight with points determined by these experiments.

If these had been disregarded, the pressures at the lower end of the curve would have been from .1 to .2 pound higher. As Regnault's values were used below 68 degrees, it seemed best to draw one smooth curve through all the points.

I have read with much interest the article by Mr. Moss in *Physical Review*, Vol. XVI., No. 6, and had intended to make a calculation to show the agreement. I can offer no explanation as to the cause of the variation cited.

---

* Author's Closure under the Rules.

No. 1021.*

THE PITOT TUBE.

BY W. B. GREGORY, NEW ORLEANS, LA.
(Member of the Society.)

1. The object of this paper is to call attention of the members of the Society to a simple, efficient instrument for measuring the velocities of fluids.

This instrument was invented by Pitot in 1730, the form first used by him is shown in Fig. 47. It consisted of a glass tube, bent at right angles, one end of which was placed in a stream of moving liquid, the end facing the direction of motion squarely with an orifice less in area than the tube. The impact due to velocity forced the liquid up the vertical portion of the tube to a height h and gave a measure of the velocity according to the law $v = \sqrt{2gh}$. A second tube open at the bottom, was added later; the two tubes being placed in a groove in a triangular prism of wood. Fig. 48 is a copy of the drawings of the tube as described by Pitot.† E is the impact opening of this tube, placed at one angle of the triangle forming the cross-section of the prism. Suitable scales FG and LI were made on the metal slide which is provided with clamps so that it could be raised or lowered as the ends of the tubes were placed at the desired depth; from these scales were obtained the difference of level of the water in the two tubes and direct readings of velocity respectively.

2. The Pitot tube was mentioned in a paper read before the French Academy of Sciences by M. Dubuat in 1753, and the suggestion offered that the tubes be made of tin, and that they contain floats carrying a rod moving in front of a scale from which the

---

* Presented at the New York meeting (December, 1903), of the American Society of Mechanical Engineers, and forming part of Volume XXV. of the *Transactions*.

† *Histoire de l'Académie des Sciences*, 1732, page 376.

heights could be read. The instrument was improved and used by Darcy and Bazin; Fig. 49 shows the form used by them. It consists of two tubes, one drawn to a fine point and pointing up stream, the other with small openings on top and bottom at the point a as shown in Fig. 49. The first of these is the impact tube and the second the pressure or static tube; the tops of the two tubes are connected so that a partial vacuum formed above the liquid in them will cause the liquid in both to rise by the same amount

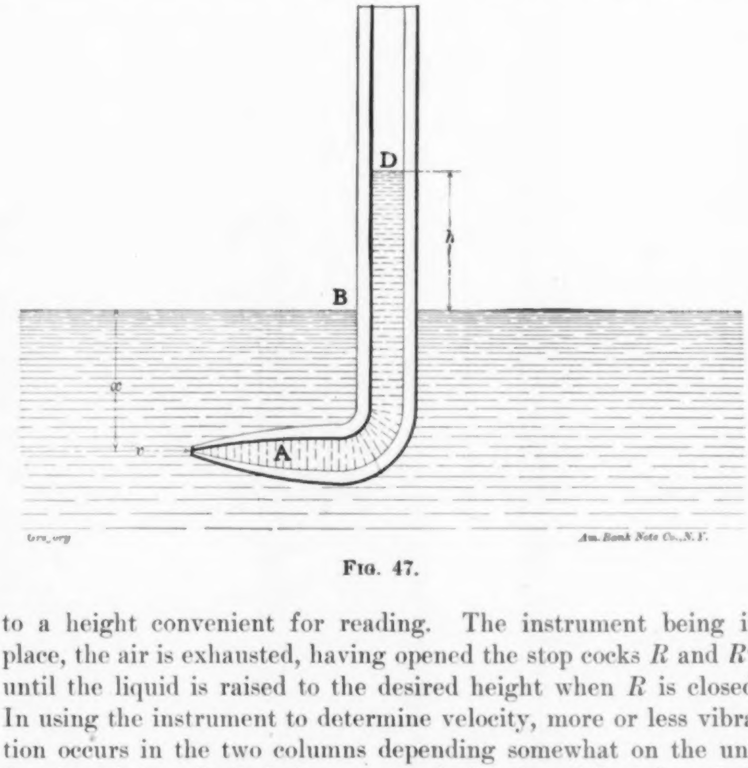


Fig. 47.

to a height convenient for reading. The instrument being in place, the air is exhausted, having opened the stop cocks R and R', until the liquid is raised to the desired height when R is closed. In using the instrument to determine velocity, more or less vibration occurs in the two columns depending somewhat on the uniformity of the velocity being measured, the size of the tubes, their openings, etc. Maximum and minimum readings may be observed and a mean computed, or a mean reading may be determined directly from the instrument. If the openings are small, the vibrations will be small; partially closing the cock R' has the effect of causing the instrument to average the readings. Finally R' may be quickly closed at the mean reading and the difference of level read with deliberation. The researches of Darcy and Bazin

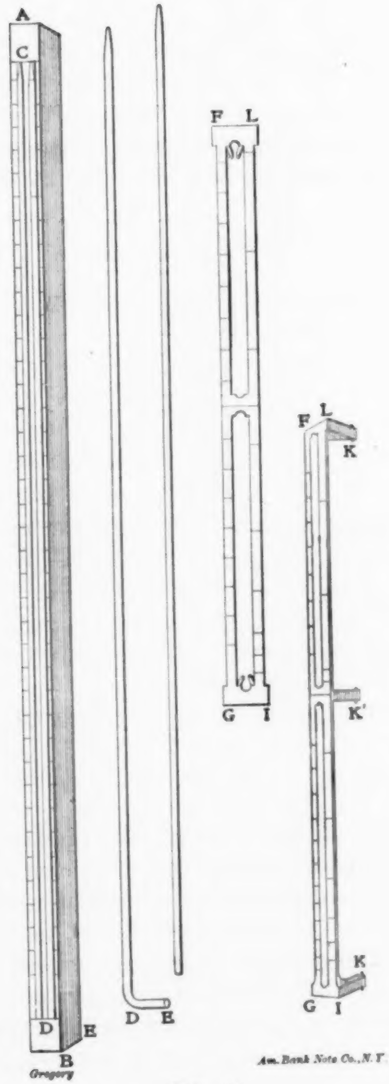


FIG. 48.

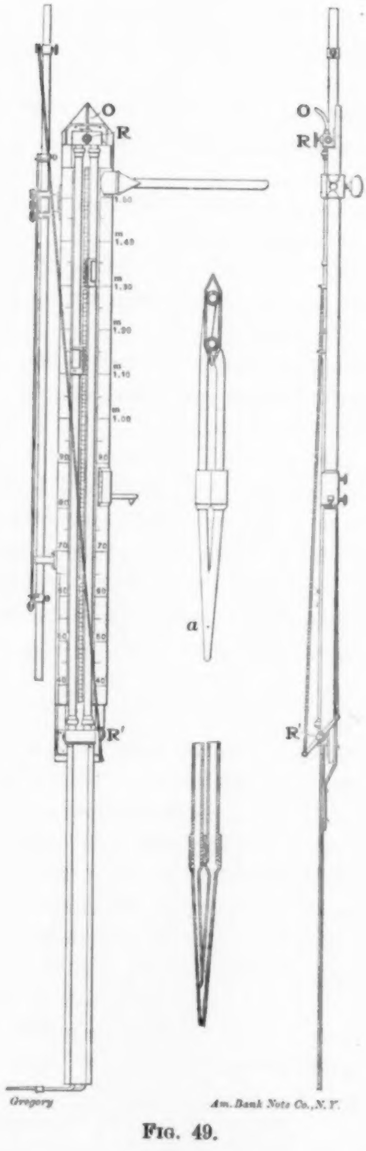


FIG. 49.

were published in 1865. They found for the tube described that in the formula $v = \varphi \sqrt{2gh}$, φ was approximately equal to unity, its value varying slightly according to the three different methods of rating which were as follows:*

(1) By placing the tube in front of a boat and drawing it through still water at different velocities. This gave a coefficient of 1.034.

(2) By observing the readings in a stream, the velocity being obtained by floats, coefficient 1.006.

(3) By readings taken in different parts of a canal in which the amount of water passing was known by measurements of its volume. Coefficient 0.993.

The first coefficient was believed to be too large, as the current may have struck the orifice at an angle, due to the water being raised in front of the bow of the boat.

3. Many others have experimented with this instrument in recent years, the instrument taking different forms, according to the ideas of the men who designed and used them. The form departed radically, in many cases from that used by Darcy and Bazin; the results obtained have been variously interpreted. Most of these instruments have had a coefficient φ less than unity in the formula $v = \varphi \sqrt{2gh}$, which transposed gives a value of h, greater than $\frac{v^2}{2g}$, the head due to velocity. This led to the belief† that the correct theoretical formula for the Pitot tube is $v = \sqrt{gh}$ and it is so stated in some text books. This belief was strengthened by the fact that in some cases the rating of the tubes gave results which corresponded reasonably well with this formula, as in the experiments of Weisbach.‡

4. A common belief also, was that the shape of the impact end, whether conically divergent, conically convergent, with thin walls or thick walls, had a marked effect on the value of the constant of a particular tube. As a matter of fact, it makes very little difference so far as accuracy of results is concerned, what the form of the instrument is or what the value of its coefficient may be if we know its value, for in any case we may write $v = C\sqrt{h}$.

* *Recherches hydrauliques, entreprises par M. H. Darcy continuées par M. H. Bazin. Extrait des Mémoires présentés par divers savants à l'Académie des Sciences de l'Institute Impérial de France*, Paris, 1865, Vol. XIX.

† See Vol. XXII. *Transactions* A. S. M. E., page 284.

‡ Church's "Mechanics of Engineering," page 752.

However the value of the coefficient must be known, in general, by careful rating, which in itself requires a large outlay of time and labor and also requires conditions which may not be within easy reach of the engineer.

5. All this diversity of opinion as to whether the correct theoretical formula was $v = \sqrt{2gh}$ or $v = \sqrt{gh}$ and the conflicting values of the coefficients which had been found, may be summed up in two questions—(1) Which formula is theoretically correct? (2) Can a tube be built which will fulfil these requirements and so not require rating?

6. The answer to these questions was given by W. M. White in a paper before the Louisiana Engineering Society in May, 1901. This paper was entitled "The Pitot Tube; Its Formula." * It is a report on experimental research of the highest order and reflects great credit on its author.

7. After a brief description of the different formulæ that have been used with the Pitot tube, a method is given of deducing its equation. This is followed by descriptions of experiments which lead to the final conclusions.

8. Mr. White first investigated the distribution of pressure in a flat circular plate on which a stream of water was allowed to fall, the plate having openings from which the pressure of the liquid above could be obtained. He found by carefully adjusting the plate so that its centre was exactly under the falling column of water that the pressure at this point was just sufficient to support a column of water equal to the height from which the water had fallen, and that the total pressure on the plate, as obtained by summing the pressures of the annular rings composing the plate, was approximately equal to twice the area of the jet times the height through which the water had fallen. These experiments were followed by others on the impinging of a falling jet of water upon nozzles of different forms. Four different nozzles were used; they are shown in Fig. 50.

9. Briefly, he found that after the nozzles were adjusted accurately to the centre of the falling stream of water, that the pressure produced was capable of sustaining a column of water almost exactly equal to the height of the source of the falling water, the very small discrepancy being due to the friction of the falling water with the air. This was true in every case, showing

* "Journal of the Association of Engineering Societies," August, 1901.

that the form of the impact tube did not influence the constant of a Pitot tube in case the opening be placed at the centre of a surface of revolution.

10. These tubes having nozzles *A*, *B*, *C* and *D* of Fig. 50 were then placed in front of a catamaran and hauled through still water in an open canal, the velocity being obtained from the time required to pass over a measured course, and the height from the

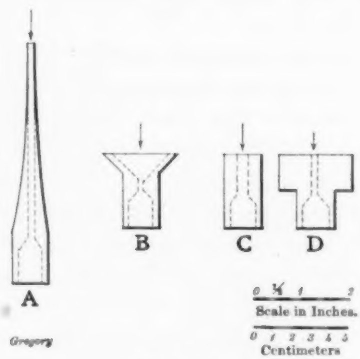


FIG. 50.

surface of the water, to which the liquid was forced, accurately measured from the surface of the canal. The constant φ to be used in the formula $v = \varphi\sqrt{2gh}$ as obtained from these experiments was 1.0053 for nozzle *D*.

11. Experiments were next conducted which showed that with a properly designed static opening the constant of a tube is practically unity. The form of static opening as used by Mr. White is similar in principle to that used by Darcy and Bazin. An ordinary $\frac{1}{4}$-inch gas pipe about five inches long was drawn to a point and a $\frac{1}{16}$-inch hole drilled through its sides about four inches back from its pointed end, in a horizontal plane. Careful calibrations of tubes having this form of static attachment showed an average constant of unity well within the error of observation. Other tubes were experimented with, in which there was a suction action on the static side; this being added to the head due to impact made the head, as read, greater than the head due to velocity or $\frac{v^2}{2g}$ and therefore gave a coefficient φ less than unity in the formula $v = \varphi\sqrt{2gh}$.

12. Mr. White sums up the results of his researches as follows:

(1) That an impact tube, whose impinging surface is one of revolution, converts velocity head into static head exactly according to the law $v = \sqrt{2gh}$ whatever the pressure of the surrounding fluid.

(2) That only pressure openings which give the true static head of water should be used in connection with the point of a Pitot tube. That is to say, that only tubes which have unity as their coefficient should be used.

(3) That Pitot tubes whose constants are unity in open canal ratings will remain unity, whatever the pressure of the liquid.

Mr. White had very few experimental data to support conclusion (3).

13. A voluminous paper * of great merit by Messrs. Williams, Hubbell and Fenkell appeared in 1902, the title is "Experiments at Detroit, Mich., on the Effect of Curvature Upon the Flow of Water in Pipes." This paper, together with its discussion, is one of the most valuable contributions to the subject of the Pitot tube, as all measurements of velocity were made with various forms of that instrument. A large portion of the work consisted in the rating of these tubes under various conditions; several different forms of the instrument were used. Among the authors' summary of results accomplished are the following:

A. "The invention of the oil differential gauge, by which it is possible to observe differences of head in closed conduits under pressure with as great a degree of precision as is attainable in open conduits with the hook-gauge."

B. "The invention of a form of Pitot tube which can be inserted in a water main without the aid of special devices, other than the tools possessed by every water department; by the aid of which a competent observer may obtain gauging with as great accuracy as has yet been attained with any other measuring device, except the graduated tank and weighing scale."

C. "The determination of a ratio .84 between the mean and maximum velocities of water flowing in closed circular conduits, under normal conditions, at ordinary velocities; whereby observations taken at the centre under such conditions, with a properly rated Pitot tube, may be relied upon to give results within 3 per cent. of correctness."

---

* "*Transactions* American Society of Civil Engineers," Vol. XLVII., April, 1902.

D. "The presentation of a series of coefficients for application to the different fluids used in the fluid differential gauges, by which the observations so taken may be conveniently reduced to equivalents in water."

E. "The demonstration of the fact that ratings of Pitot tubes made by dragging these instruments through still water in open troughs do not conform, within any reasonable limits to those obtained when the instrument is stationary in moving water in a closed conduit."

F. "The demonstration that Pitot tubes must have their coefficients determined, whether they consist of point opening alone or both point and pressure openings."

G. "The demonstration that under some conditions, in straight pipe there is a difference of pressure at different points around the circumference of the same cross-section."

H. "The derivation of the ellipse as the approximate form of the normal curve of velocities in straight circular pipes."

14. As was to be expected the discussion of the paper brought forth many valuable contributions on both theoretical and experimental sides. Among those who discussed the Pitot tube are to be found the names of many of the highest authorities on hydraulics. The paper with its complete discussion must be read to be appreciated; only a few points, which have a direct bearing on the subject in hand, will be mentioned.

15. In the final discussion of this paper by its authors, after carefully weighing the contributions and criticisms of the various men who discussed it, we find the following:

"The weir, the current meter, the water meter, and the nozzle are the only other devices to be compared with the Pitot tube, and the results obtained with the first three are rarely if ever better than those presented herein, and two of them are not directly applicable to the flow of water in pipes. The nozzle may perhaps give somewhat better results, but it is limited to comparatively small streams. It seems therefore that the writer's conclusion, that the Pitot tube, properly rated in the hands of a skilled observer, is as accurate as any other device for measuring water in a closed conduit, except the scale and tank, is fairly sustained."

16. "The writers, therefore, believe that conclusion *E* is sustained, and that an instrument should be rated under conditions as nearly as may be similar to those under which it is to be used, and that dragging an instrument through still water does not conform

with this condition when the apparatus is to be used to measure running water, either in closed or open channels. In the light of present knowledge, if instrument can only be rated by dragging, the writers would prefer the coefficients established by dragging in opposite directions in running water to those obtained by dragging in still water."

17. "In view of the data presented by Mr. White and of the experiments of Messrs. Saph and Schroder, and Adams and Wilson, wherein instruments of certain forms were used with ring piezometers, as well as the experiments of the writers regarding obliquity it seems to be demonstrated satisfactorily that conclusion *F* was too sweeping, and the writers agree as pointed out by Mr. Ferris, a proper form of point opening will have a coefficient of unity by the formula $h = \frac{v^2}{2g}$ when combined with a ring piezometer under normal or nearly normal flow."

18. Again in discussing result *G* they say, under head of "Pressure variation," "Conclusion *G*, that the pressure in a straight pipe is not always the same at all points around the circumference of the cross-section, is based first upon the experimental evidence in the 16-inch pipe investigation. No discussion has been submitted containing direct additional evidence upon this particular point. All the Pitot-tube work shows that the components of velocity parallel to the axis of the pipe vary from point to point throughout the cross-section, and that changes of that component of velocity take place at each individual point under distorting influences. If these components represent or are proportional to, the total velocity in each part of the fluid, it follows from the Law of Conservation of Energy, since all the water in the cross-section was originally started in its course under the influence of the same head, that if velocity head be increased, pressure must be decreased, unless the elevation of the stream above datum be decreased, whether a single particle or the whole stream be considered, and therefore, that the pressure must vary throughout the cross-section, a conclusion which is confirmed by the most obvious interpretations of a large number of experiments. The experiments of Messrs. Adams and Wilson, and Saph and Schroder, as well as some of those of the writers, prove pretty conclusively that a properly formed point combined with a ring piezometer gives readings the summation of the deducted velocities of which amount to the true discharge, on the theory used by the writers

that $h = \frac{v^2}{2g}$, without introducing any coefficient, as has been demonstrated in the case of a jet and in open channels by Mr. White. This being true, the acceptance of varying pressure throughout the cross-section, which pressure varies inversely as the velocity leads to some complications, if one undertakes to maintain that the point effect is $\frac{v^2}{2g}$. Another interesting circumstance is the reading given by two conical points, one receiving the impact of a current in a closed pipe and the other directed down stream. Theoretically it might be expected that the difference between these readings would be $\frac{2v^2}{2g}$ and this alleged fact is the basis of the claim for a patent granted for such device some years ago, but, experimentally, or practically, nothing of the kind happens with his apparatus."

19. "Regarding the question of the proper formula for the Pitot tube, the writers are not yet entirely satisfied, and while fully appreciating Mr. Frizzell's effort to clear the matter up, they yet prefer to leave the question where they left it before, that, practically, it makes little difference in the reduction of results whether a tube reading actually is $\frac{v^2}{2g}$ or $\frac{v^2}{g}$. The discussion presented by Mr. Seddon gives excellent reason to believe that there is a large account of internal forces to be balanced before the laws of flow will be fully understood, and until these forces have been more fully investigated, too rigid laws had best not be laid down."

20. In the discussion of this paper there is a contribution by Edward S. Cole in which he describes a series of independent experiments conducted by him. A device, which he has named a Photo-Pitometer, was invented by him and used to obtain a continuous record of the readings of a Pitot tube, extending over a period of time. To accomplish this result clockwork, lamp and photographic paper were used. For descriptions see original paper.*

21. It appeared that the oil differential gauge had been invented and used by independent investigators in at least five different localities and that in one case had it been patented.

22. In quoting from an extensive work like the one just re-

* Vol. XLVII. *Transactions* American Society of Civil Engineers, April, 1902, page 275.

ferred to, there is some danger of conveying a wrong impression by merely giving those parts to which it is desired to direct attention. The writer wishes to disclaim any such intention and to state that the particular parts quoted have been selected because they have a direct bearing on the theory and use of the Pitot tube and are along the lines on which it is wished to discuss some further results which possibly throw additional light on the subject.

23. During the fall and winter of 1902 the writer was consulting engineer in a series of tests of the Hydraulic Dredges of the Mississippi River Commission.

24. By means of these dredges a navigable channel is maintained in the river during low water. Large centrifugal pumps, driven by direct connected steam engines and capable of maintaining a mean velocity of from 15 to 22 feet per second in the discharge pipes, are used as dredge pumps.

25. The series of tests, which extended over several months, included efficiency tests of boilers, engines and dredge pumps besides tests of accessory machinery such as "jet pumps" used to loosen the material to be removed in dredging. Some of these "jet pumps" were of the reciprocating steam pump type while others were centrifugal pumps driven by direct connected engines.

26. A complete report in detail of these tests can be found in the Report of the Chief of Engineers of the U. S. Army for 1903. It is in the report of F. B. Maltby, U. S. Assistant Engineer, Member of the American Society of Civil Engineers, Superintendent of Dredging Operations for the Mississippi River Commission, and forms a part of the report of Captain W. B. Ladue, Corps of Engineers of the U. S. Army, Secretary of the Mississippi River Commission.

27. The general method of these tests was outlined by the writer who was present while some of them were made. The details were worked out and the tests conducted by Mr. Maltby, who deserves much credit for a work of great magnitude in which success was attained only by untiring labor and careful attention to details.

28. The results are all of great interest, however. In the discussion which follows reference will only be made to those parts of the work which have a direct bearing on the subject in hand, viz.: the Pitot tube.

29. The problem of measuring the mean velocity of the water in the discharge pipes was of prime importance as the amount of

water pumped and the efficiencies of pumps both depended on the accuracy of measuring this velocity. The quantity of water in some cases exceeded 120 cubic feet per second. The diameter of discharge pipes were approximately 32 inches in some cases and 34 inches in others. Seven dredges were tested; the method used must be applicable to all. Had a weir been used it would have been available for only a short time, owing to a rapidly rising river. A Venturi Meter would have introduced a resistance in the discharge pipe, and rendered the conditions quite dissimilar to those of ordinary running, which were desired for these tests. The velocity to be measured at the centre of the pipe was in some cases as great as 25 feet per second. The loss of head at the throat of a Venturi Meter, while small at low velocities, would be enormous at the velocities used even if a meter considerably larger than the size of pipe were used.

30. A measuring barge into which the water could be deflected for a brief interval of time, and measured, had been used in testing these dredges in 1897. A complicated mechanism was required to deflect the stream of water and the short interval in which the discharge was measured, together with the uncertainty of getting the time exactly were the objections to this scheme.

31. All these methods, besides being expensive, are difficult to apply, partly because the discharge pipes of some dredges are half submerged in the river, the pontoons being so designed—while in other cases the pontoons support the pipes 30 inches above the water.

32. The Pitot tube method was recommended and used and it is believed that the results obtained are as accurate as could be had by any of the other methods suggested; besides ease of application, cheapness, and the fact that the pumps could be tested under conditions exactly similar to ordinary running, are all in favor of this method. The tubes used were designed by Mr. Maltby, they were made in the machine shops aboard the dredges as were the gauges which were used in connection with them. Nine tubes were constructed, but all except Nos. 1, 3, 8 and 9 were discarded because of apparent defects.

Description of Pitot Tubes.

33. Tube No. 1 consists of two pieces of $\frac{1}{4}$-inch brass tubing, enclosed in a pipe $1\frac{5}{16}$ inches outside diameter. One of the small

brass tubes is bent at the end below the end of the outside enclosing pipe, through 90 degrees and the plane of the opening made truly parallel with the upright pipe to form the impact opening. The other brass tube is brazed to a solid brass piece, circular in section and placed at right angles to the tube, the upstream end has a sharp point which is even with the opening of the impact tube and below it; it has a $\frac{1}{16}$-inch hole drilled on each side connecting with the interior of the upright tube, and forms the static opening: ordinary $\frac{1}{4}$-inch air cocks form the upper side of these tubes; $\frac{1}{4}$-inch rubber tubing about 4 feet long is used to connect these air cocks to the gauges.

34. Lead or cement is used to hold the small tubes in place in the outer tube. A stuffing box was used through which the outside tube slides; it is screwed into a hole tapped in the top of the pipe with a $1\frac{1}{2}$-inch standard pipe tap. A handle at the top of the tube carefully set at right angles to the plane of the impact opening was used to move the tube vertically from top to bottom of the pipe in traversing and enables the tube to be kept in alignment with the axis of the pipe.

35. Tube No. 3 is similar to No. 1, with the exception that the impact point is below the static point.

36. Tube No. 8, and in fact all the different tubes have the same size vertical outside pipe to permit the use of the same stuffing boxes. At its lower end it is joined at right angles to a 1-inch pipe about 18 inches long, drawn down to a point of the proper size to admit a $\frac{1}{4}$-inch brass tube which is brazed fast into it forming the impact opening. The small tube runs inside the larger pipe to the top, where it terminates in a $\frac{1}{4}$-inch air cock. The horizontal pipe has three $\frac{1}{16}$-inch holes drilled in its side, forming the static openings; the interior of the pipe is connected through the upright and one handle to a $\frac{1}{4}$-inch air cock.

37. Tube No. 9 was made as nearly like No. 8 as could be done by a skilled mechanic.

It was very soon apparent that the tubes Nos. 1 and 3 did not have a constant φ of unity in the formula $v = \varphi \sqrt{2gh}$.

38. It was believed that tubes No. 8 and 9 would give a value of unity, but to prove this point was another matter. It has been shown conclusively by Mr. White that an impact point similar to that of tubes Nos. 8 and 9 would register the level corresponding to a given velocity according to the law $h = \frac{v^2}{2g}$ when the

tube was dragged through still water in an open canal. Messrs. Williams, Hubbell and Fenkell, as we have seen, agree that a "point combined with a ring piezometer gives readings, the summation of the deduced velocities of which amount to the true discharge" on the theory used by them "that $h = \frac{v^2}{2g}$ without introducing a coefficient. If then it can be shown that these tubes when calibrated in running water in an open channel give a constant φ of unity in the formula $v = \varphi\sqrt{2gh}$ and also shown that the static readings of the tubes correspond to the readings of the ring piezometer, it would seem that the question of the value of the constant may be settled in this way. Furthermore it has been claimed that the constant of a tube rated in an open channel is different from that obtained by rating in a closed conduit under pressure. If this is true, tubes Nos. 8 and 9 being exactly alike and having the same constant when used simultaneously to traverse two sections of a discharge pipe at different points where the mean velocity of the water is necessarily the same, but the static pressure very different will show decided differences in their traverse and in the consequent mean velocity. We shall see what was obtained by the experiments.

39. Tube No. 9 was rated in running water having a velocity of about $3\frac{1}{2}$ feet per second by comparing it with floats; about 80 floats were run and 600 observations made.

40. The tube was suspended in the Mississippi River with the point about 12 inches below the surface, facing the current. An inverted U-tube of glass was used, the air being partially exhausted from the top to bring the water surfaces to a convenient height for reading.

41. The floats were 24 inches long; the time was observed in passing over a course 40 feet long, the tube being placed about one-third the distance from the upper end of the course.

42. In the table which follows the velocity by floats is in each case the mean of ten observatons, and that given for the Pitot tube is the mean of about 75 observations:

No.	Velocity of Floats.	Velocity by Tube.
1	3.316	3.420
2	3.284	3.279
3	3.517	3.393
4	3.419	3.358
5	3.405	3.502
6	3.212	3.228
7	3.362	3.354
8	3.332	3.218
Means	3.356	3.344

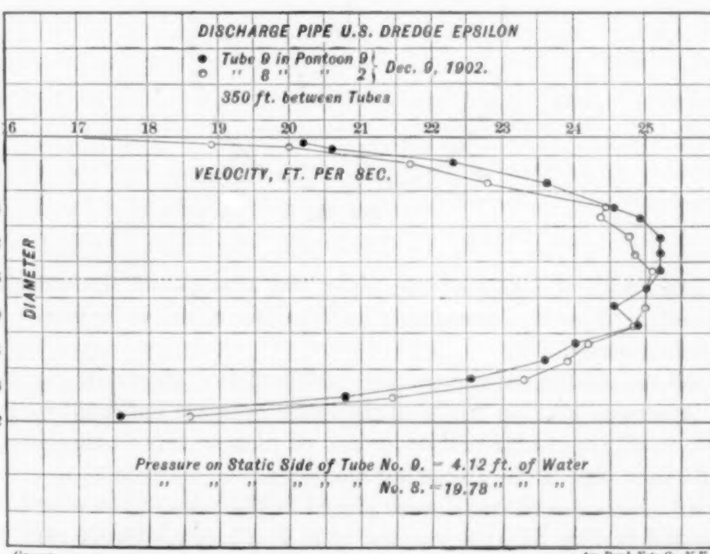
The means show a coefficient of unity within about one-third of 1 per cent.

43. Tubes 8 and 9 were placed in the same discharge pipe 50 feet apart, and three sets of observations made; the tubes were then changed about and three sets of observations taken again, with the following results. Each set is the mean of ten readings.

	Velocity by Tube No. 8.	Velocity by Tube No. 9.
	25.37	24.50
	24.16	25 60
	25.77	25.14
	Reversed Positions.	
	25.36	25.19
	25.37	24.84
	25.55	25.02
Means........	25.263	25.048

The means agree within about eight-tenths of 1 per cent.

44. To test the accuracy of velocity measurements by these two tubes and determine whether or not they were affected by static



Gregory Am. Bank Note Co., N.Y.

FIG. 51.

pressure, tube 8 was placed in the discharge pipe of the second pontoon of the U. S. Dredge Epsilon, where the average static pressure was 19.78 feet of water, and tube No. 9 was placed in

the ninth pontoon, 350 feet from tube No. 8, and where the average pressure was 4.12 feet of water. Traverses were made by taking observations simultaneously at different points across the pipe. The mean velocity determined by tube No. 8 from 170 observations, was 22.321 feet per second, while that determined by tube No. 9 was 22.351 feet per second, a difference of one-tenth of 1 per cent. The mean velocity was obtained from the traverses in a vertical plane, by dividing the area of the pipe into ten equal areas the boundaries of these areas being circles con-

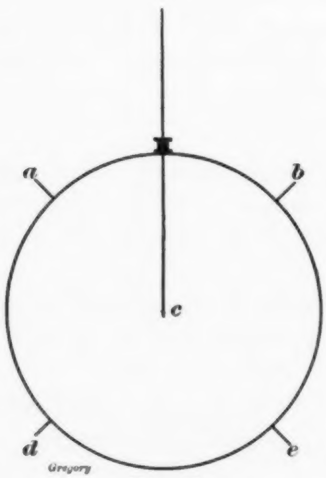


FIG. 52.

centric with the axis of the pipe; the average of the velocities found in these ten areas was used as the mean velocity for the whole cross-section.

45. A plotting of the traverse referred to is shown in Fig. 51.

A comparison was made of the static pressure indicated by tube No. 9 with that obtained by means of piezometers in the sides of a 32-inch pipe in the following manner:

46. Four $\frac{1}{4}$-inch cocks were placed in the sides of the pipe as indicated in Fig. 52 at the points of *a*, *b*, *d* and *e*. Great care was taken to have the axis of these cocks exactly normal to the axis of the pipe; their ends projected inside the pipe slightly, and after being screwed into place these ends were filed off carefully to present a perfectly smooth surface, flush with the inside of the pipe. When each of the piezometers was connected in turn with

each other through a differential gauge no difference in pressure could be observed. Tube No. 9 was then inserted into the same vertical section as the piezometers, the point of the tube being at the centre of the pipe. The static side being connected successively with each piezometer, through a differential gauge, no differences of pressure could be observed. The pressure of the four piezometers and the static side of the tube was exactly the same.

47. In Fig. 53 are shown four traverses of the discharge pipe of

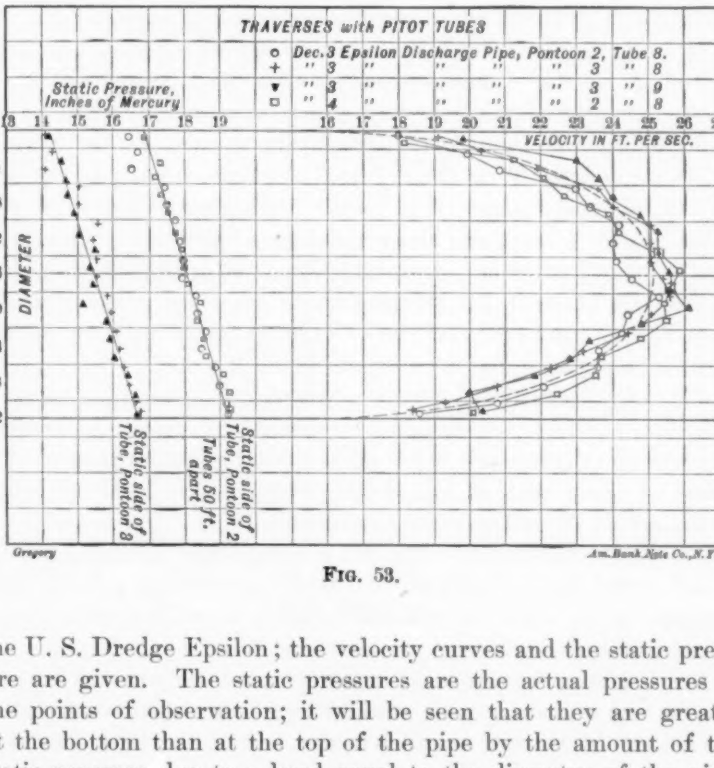


FIG. 53.

the U. S. Dredge Epsilon; the velocity curves and the static pressure are given. The static pressures are the actual pressures at the points of observation; it will be seen that they are greater at the bottom than at the top of the pipe by the amount of the static pressure due to a head equal to the diameter of the pipe. Irregularities are no doubt due, in part, to the practical impossibility of keeping steam pressure and consequently the revolutions of the engines and pumps absolutely constant. Each point plotted is in general the average of ten readings, considerable time being required to complete a traverse, as much as an hour in some cases.

48. These readings were obtained by using two ordinary U-gauges and observing impact and static pressures separately. The work required by this method is much greater than when a differential gauge is used and the difference of level of the mercury columns only is taken. The latter gives readings from which velocity can be computed, but does not show the way in which the pressure is distributed across the pipe. In case separate static and impact readings are desired, two U-gauges must be used, and while one side of the mercury column in the U-tube is open to the atmosphere, the side connected to the tube has a solid column of water resting on it. Great care has to be exercised to expel all air, and to be sure that the water column is solid before readings are taken. Knowing all the dimensions of gauges, and consequently the relative height of the mercury columns, with reference to the point being investigated, the absolute pressure of that point is easily computed. With the differential gauge, when velocity only is wanted, the two ends of a U-tube containing mercury are connected to the impact and static openings of the tube. Care must also be exercised in getting solid columns of water on both sides in this case before reading; the true difference of pressure in this case, is that due to the differences of level of the mercury, diminished by a pressure due to an equal height of water.

49. After the ratio of velocity at the centre of the pipe, to mean velocity, had been determined by repeated traverses, observations were taken at the center only, while testing the main pumps. The value of this contstant varied somewhat, due to local conditions at the points where traverses were made.

50. Referring again to Fig. 53 a semi-ellipse has been drawn, using dotted lines; it will be seen that it represents a fairly good average of the observed velocities; this is more apparent when a larger number of traverses are plotted.

51. It is seen from these results that the static pressure does not vary across the section of a straight pipe in which there are great differences of velocity parallel to the axis of the pipe, except as affected by gravity. This shows conclusively that the sum of static and velocity heads for various points in the same cross-section of a straight pipe, is not a constant quantity if we understand by velocity head, that due to the velocity parallel to the axis of the pipe. This ought not to destroy our belief in the Law of the Conservation of Energy. All the energy possessed by a particle of water at any point of the cross-section of a straight

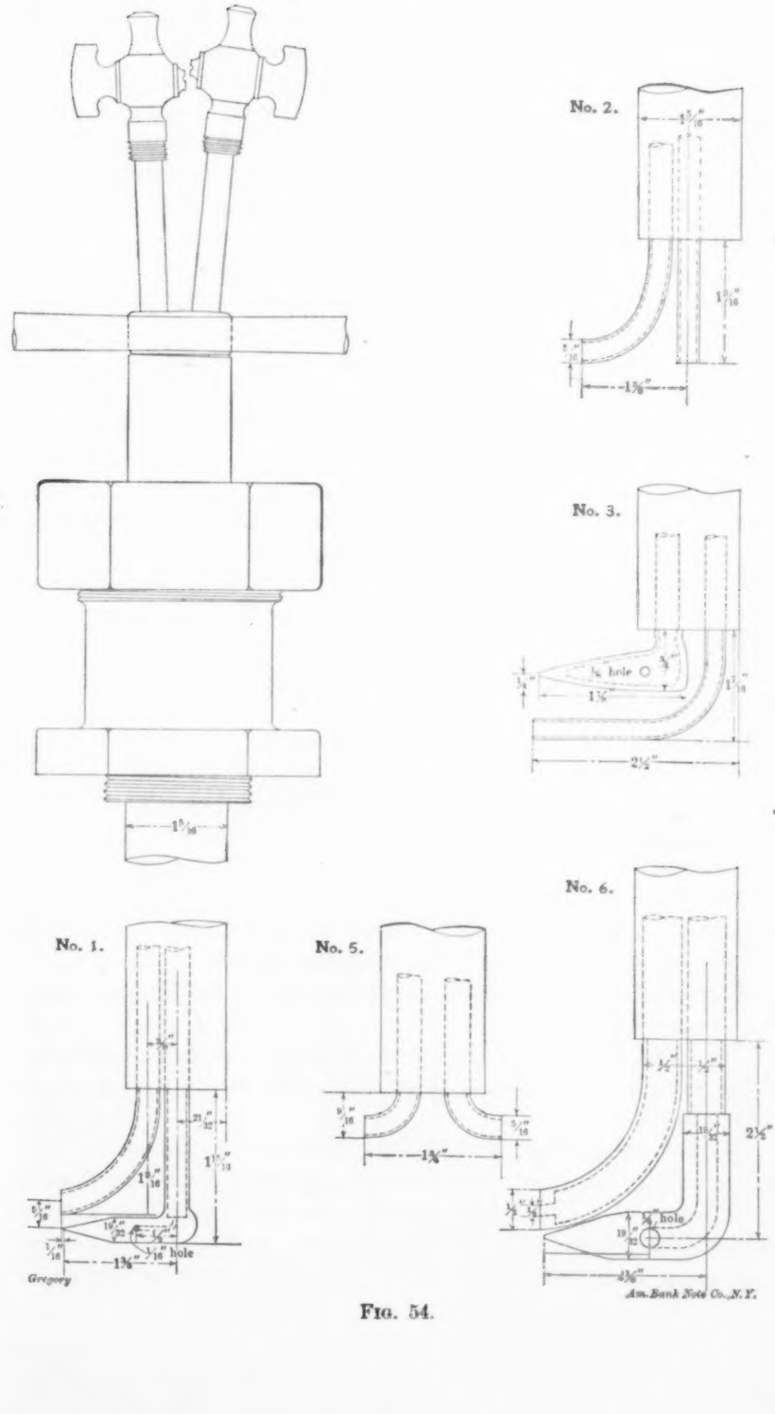


FIG. 54.

pipe is either energy or position, pressure or motion. Since the pressure energy is constant across a given section and the energy of motion parallel to the axis of the pipe varies, it follows that there must be energy of motion other than this. Unfortunately we have no way of measuring the velocity of the whirl of the particles of water at various points in the cross-section. If this could be done, undoubtedly it would be found that the energy due to velocity of whirl is greatest at the walls and least at the centre of the pipe, and that the energy possessed by a particle of fluid at one point in the cross-section is equal to that possessed by any other particle in the same section.

52. It is impossible to convert this energy due to velocity of whirl into useful work; of course it is finally converted into heat.

Tube Nos. 1 and 3 were used in important tests; they were rated by comparing with tube No. 8 by placing them in the same discharge pipe 50 feet apart; three sets of observations of ten each were taken, then the positions of the tubes reversed and three more sets taken. The coefficients of tube No. 1 was found to be .930 and of tube No. 3 .8915. These values were used in reducing observations taken by these tubes.

53. The cut shows the discarded tubes.

Tube No. 2 had an impact opening similar to tubes 1 and 3. The static opening was a vertical brass tube, with lower end cut off squarely, the axis of the tube being at right angles to the current measured. This tube gave the greatest amount of suction at the static opening, found in any of the tubes. The suction was so great that when the impact side was made to face down stream the reading of the impact side was still greater than that of the static side.

Tube No. 4 was tube No. 2 with the lower end of the static tube plugged up and with openings on the side.

Tube No. 5 had impact and pressure points similar, but pointing in opposite directions.

Tube No. 6 was very similar to tube No. 1, except that large tubes were used, the impact being filled with a plug having a hole $\frac{1}{8}$-inch in diameter.

54. No rating was made to determine the coefficient of these tubes, except No. 6, as a search was being made for a tube having a coefficient of unity, and it was soon evident that this was not the case with any tubes, except Nos. 8 and 9.

55. The use of the Pitot tube is by no means confined to meas-

uring the velocity of water and liquids, but has been used to measure the velocity of air and gases. Professor Carpenter in his work entitled "Heating and Ventilating Buildings," page 41, describes a tube to be used to measure the velocity of air. In the same volume, page 45, reference is made to the Prussian Mining Commission which, in 1884, by means of a large gasholder which contained 70,000 cubic feet, investigated several questions relating

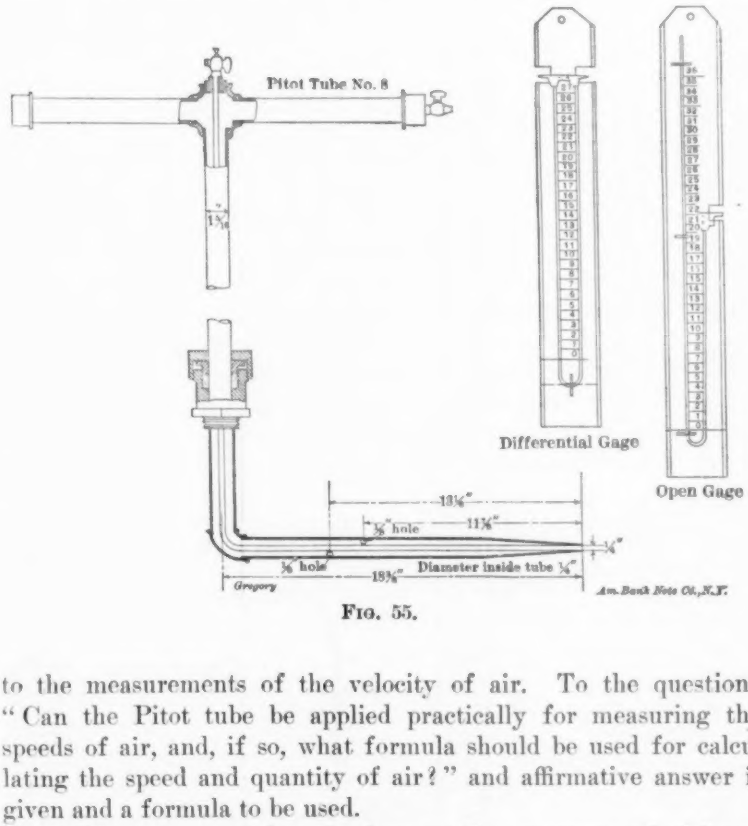


FIG. 55.

to the measurements of the velocity of air. To the question: "Can the Pitot tube be applied practically for measuring the speeds of air, and, if so, what formula should be used for calculating the speed and quantity of air?" and affirmative answer is given and a formula to be used.

56. The instrument has also been used to measure velocities of air in a series of tests of the greatest refinement conducted by Capt. D. W. Taylor, U. S. N., at the Experimental Model Basin, Washington Navy Yard. Captain Taylor used several tubes in the same cross-section so arranged that the mean velocity was the arithmetical mean of the several velocities given by the tubes.

57. Pitot tubes were adopted for these tests after extensive ex-

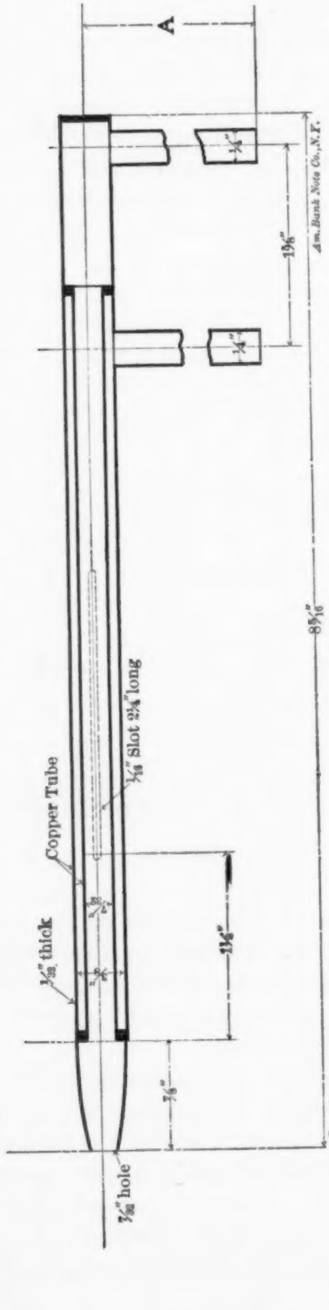

A
¼"
1⅜"
¼"
Am. Bank Note Co., N.Y.
8 5/16"
1/16" Slot 2¼" long
Copper Tube
1/32" thick
1½"
⅞"
7/32" hole
Gregory

FIG. 56.

periments with anometers. Static and impact pressures were obtained separately by means of instruments of great precision. The form of tube used is shown in Fig. 56 which gives a section through the centre of the tube. The static openings consist of two slots on opposite sides of the tube; one of these is shown in the cut. It was assumed that the constant φ, for this tube, is unity in the formula $v = \varphi\sqrt{2gh}$.

58. An account of this work will be published in the near future by Captain Taylor.

59. The three final conclusions of Mr. White, in the discussion of his paper were:

(1) "That an impact tube, whose impinging surface is one of revolution, converts velocity head into static head exactly according to the law $v = \sqrt{2gh}$, whatever the pressure of the surrounding fluid."

(2) "That only pressure openings which give the true static head of water should be used in connection with the point of a Pitot tube. That is to say, that only tubes which have unity as their coefficient should be used."

(3) "That Pitot tubes whose constants are unity in open canal ratings will remain unity, whatever the pressure of the liquid."

60. Experimental data confirming the first have been given.

The second is obvious on the basis of economy and general desirability.

The third has been confirmed by the experiments with tubes Nos. 8 and 9.

61. Finally, the writer believes that the Pitot tube is an instrument by means of which fluid measurements, whether of liquids or gases, may be made with as great accuracy as with any of the ordinary devices used. It is inexpensive and easy of application —is capable of being used for both high and low velocities, and may be used to measure velocity of flow in pipes without materially changing the normal condition of flow; furthermore, it is especially suited to many cases where other devices would be impossible of application. If properly constructed it requires no rating.

62. These facts should make the Pitot tube a popular instrument with engineers who have tests to conduct of pumps, turbines and blowers. To those interested in the complicated problems of hydraulics this instrument offers advantages not possessed by any other device.

SUPPLEMENTAL BY PROF. S. W. ROBINSON, COLUMBUS, O.

The Pitot Tube velocity instrument is one so exact, so simple, so inexpensive, and yet so easy of application, that too much cannot be said in its favor. Mr. Gregory's paper as well as several others, in which the Pitot Tube plays an important part, indicate that the excellent qualities of that instrument are at last being generally recognized. It is probably the most nearly perfect instrument yet discovered for the measurement of velocities of fluids and it is equally applicable to both gases and liquids. It applies to liquids in open channels, or to all fluids from orifices or through conduits, at low or high pressures, and gives results with remarkable precision. Thus there is no necessity for tedious calibration or for the use of complicated coefficients. Its simple construction admits of its being made of glass tubing and rubber hose.

Being thrown by these reflections into a reminiscent mood, the writer recalls that in 1873 he conducted a series of experiments with the Pitot Tube instrument to measure the velocity of flow of *air* issuing from an orifice, and to determine the dimensions of the jet. These tests may be of some interest because they are believed to have been the *first* applications of the instrument in question to the flow of gases. (See Van Nostrand's "Engineering Magazine," 1886, page 91.) The instrument used at that time was made from glass drawn down to a fine tube end for the Pitot tip, and so arranged that it could be moved from side to side or lengthwise of the jet. Thus the precise form of the longitudinal or cross section of the jet, and its velocity at any point, were studied. (See Van Nostrand's Magazine, as above.) It was shown at that time that if the tip of the instrument was placed through the orifice to the inside of the tank and then withdrawn along the middle of the jet that the pressure shown by the instrument was always the same till the *vena contracta* was passed; thus illustrating the fact that, as the internal pressure of a particle of fluid at the tip diminishes, the stored energy increases, and that, as the potential energy due to pressure falls, the actual energy due to motion rises, the sum being constant. These experiments also showed that the Pitot Tube "will exactly indicate the pressure or head due to velocity when the statical pressure is eliminated as is done in the double tip." (See illus-

trated test and proof in Van Nostrand's "Engineering Magazine" for 1886, page 94, figure 5.)

Although the writer made occasional use of the Pitot Tube instrument at other times both in the flow of gases from gas wells and water (as in 1877, the flow of the Sangamon River described in "Van Nostrand's Magazine," of 1878, page 258, and in 1893, the Flow of Water from the Castalia Springs in Ohio), yet his most important experiments were made in 1885, when he was called upon by the State Geologist of Ohio to devise a system for measuring the flow of gas wells, then being drilled in considerable numbers in that State. (See Van Nostrand's "Engineering Magazine," Volume 35, page 89, and also "Geological Survey of Ohio," Vol. 6 for 1888, page 548.) In working on this matter a careful series of tests was made in the laboratories of the Ohio State University, with which institution he was then connected. In these tests both single and double tips were used in connection with a receiver from which air issued from different-sized openings. As in the air-tests of 1873, the pressures were studied with tips within the receiver and at a distance of from one to two diameters outside of the orifices, as well as at intermediate points. It was then concluded that the Pitot Tube Instrument is a thoroughly reliable one for determining the pressure or dynamic head to which the velocity is due, and that the original notions of Pitot were correct, including that of the value of the coefficient of correction being unity, a point that all investigators have accepted as true, certainly after making a few experiments. Indeed it is wonderful how the simplest Pitot Tube Instrument will produce perfectly accurate results. This simplest of instruments is easily made with care, the essentials being all confined to the very tip ends. That is: a good Pitot tip is preferably of cylindric form for some distance back from the tip end, and preferably reamed out to a sharp edge all around, the latter being square with the cylindric body of the tip end. It is best a body of revolution, six to ten times as long as its diameter. Applied to a long pipe with a running fluid within, a second tip is preferably introduced, which tip may be formed like the other except to plug up the forward end hemispherically and cut a hole directly through square, and at about half the length. Otherwise, though the two tips as above are the most reliable, a double tip may be employed as shown in a cut in the report of the Geological Survey of Ohio for 1890, page 281.

Another interesting point brought out in these experiments is that relating to the temperatures of gases. From theoretical grounds it might be expected that in the impact of gases against the open end of a tube the temperature and density would be restored to that of the receiver as well as the pressure. For investigating this matter an "encased thermometer" was used. (See cut in Van Nostrand's "Engineering Magazine," Vol. 35, page 100.) This thermometer when properly exposed to the jet would retain a constant temperature equal for frictionless orifices to that of the receiver, while a naked thermometer similarly exposed would drop from twenty to twenty-five degrees, partly due to expansion and partly due to the impact against the thermometer.

So finally it was concluded that "when a fluid flows from a higher to a lower pressure through a frictionless orifice, the por-

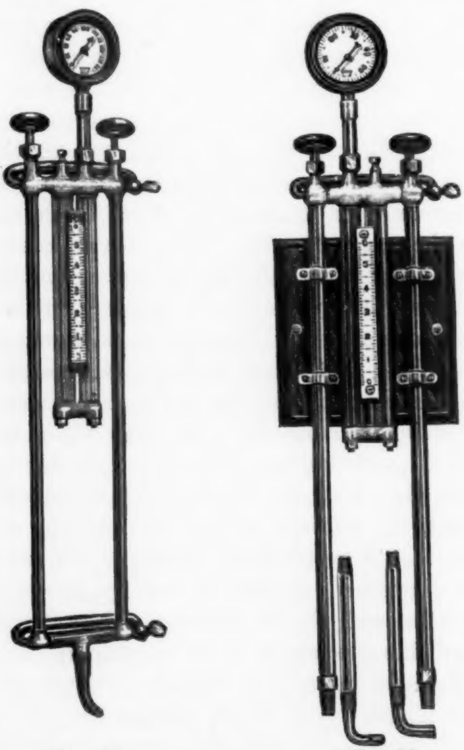


Fig. 57. Fig. 58.

Patented August 23, 1892—481,310.

tion caught in a cup mouth will be restored to its original condition as to pressure, temperature and density." This truth follows from the fact that in gases p.v.t. = a constant.

Having decided to adopt the Pitot Tube Instrument for the measurement of gas wells, the writer designed and patented in 1892 the two instruments shown in the accompanying cuts taken from the pamphlet sent out with the instrument (one for permanent installation, and one for changing from place to place, as in expert work), and brought out formulas and tables for their use in open wells or in pipe lines under great ranges of pressures. These instruments and tables have since come into very general use and the results from them are very confidently accepted. That this confidence is not misplaced may be seen from the following table of Pitot Meter results, which when compared with other meter results show very close agreements.

TABLE OF RESULTS OF NATURAL GAS PITOT TUBE METER MEASUREMENTS COMPARED WITH VARIOUS SIMULTANEOUS OTHER METER MEASUREMENTS.

Diameter of Pipe Line, in inches.	Gage Pressure in Pipe Line.	Cubic Feet per Hour.	
		By Pitot Tube.	By Gas Meter.
6	13.6 ounces	12,005	11,970
3	10.4 "	4,025	4,430
6	9.2 "	15,843	14,933
6	9.2 "	15,336	14,933
6	7 pounds	29,702	28,900
6	7 "	34,590	33,000
6	7 "	29,460	32,032
6	19.2 ounces	20,611	20,930
6	19.2 "	19,175	19,063
6	19.2 "	17,733	17,990
3	8.8 "	3,924	4,307
6	7.0 "	13,325	12,720
6	13.3 "	12,797	12,310
	Means	17,578	17,500
10	11.5 pounds	159,100	142,550
10	12.2 "	150,000	140,200
10	13.3 "	163,700	162,950
10	24.5 "	244,000	247,000
10	20.5 "	200,700	197,800
10	23.0 "	167,800	169,900
10	16.2 "	188,000	165,900
10⅝	21.0 "	1,517,350	1,516,320
10	37.0 "	1,123,000	1,102,000

10 measured on 10" line		12,841,281
8 branch from above	7,898,921	
6 2d branch	4,962,813	
Sum of two branches	———	12,861,734

No. 1022.*

CONSTRUCTION AND EFFICIENCY OF A FLEMING FOUR-VALVE ENGINE DIRECTLY CONNECTED TO 400 *KW. GENERATOR.*

BY BENJAMIN T. ALLEN, HARRISBURG, PA.
(Member of the Society.)

1. The purpose of this paper is to describe the general construction of a new type of four-valve stationary engine, and the results obtained from it. The efficiency of this type seems to exceed that obtained from much more elaborate constructions.

The machine was designed to effect not only the highest efficiency, so far produced, but also to attain this result under wide variations of load, which latter condition is usual in the majority of cases.

Before considering in detail the performance of the medium speed compound four-valve engine, which is the subject of this paper, it will not be out of place to give a brief explanation of the motives leading to its final development, and the methods used in its construction.

2. The object in view in the production of this engine, was the combination of the advantages of the most economical slow speed Corliss engines, with the very desirable features of compactness, better rotative speed, closer regulation and the more efficient methods of lubrication, possessed by the high-speed automatic engine.

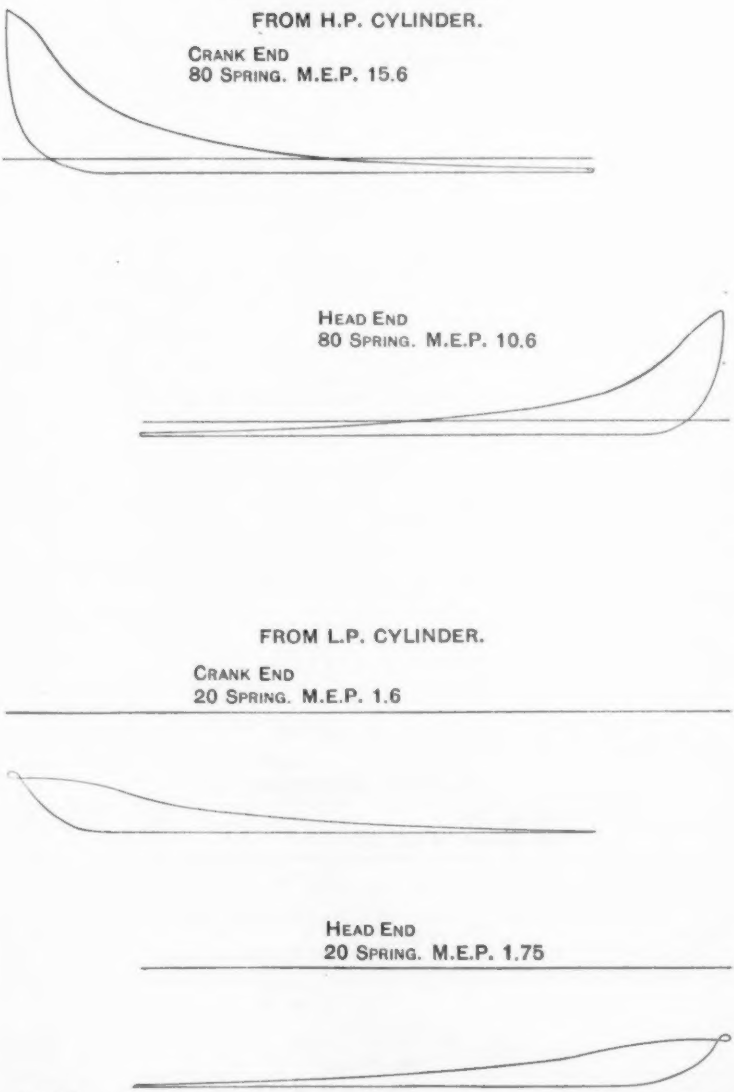
3. In order to realize the economy of steam consumption of the best Corliss engine practice, the sharp cut-off and more perfect steam distribution, attained by the use of vacuum dash pots and other accelerating devices common to Corliss engines, was at first considered absolutely essential and in the earlier stages of the development the best forms of these devices obtainable were used.

4. It was found that with an improved form of detachable cut-

* Presented at the New York meeting (December, 1903) of the American Society of Mechanical Engineers, and forming part of Volume XXV. of the *Transactions.*

SPECIMEN CARDS.

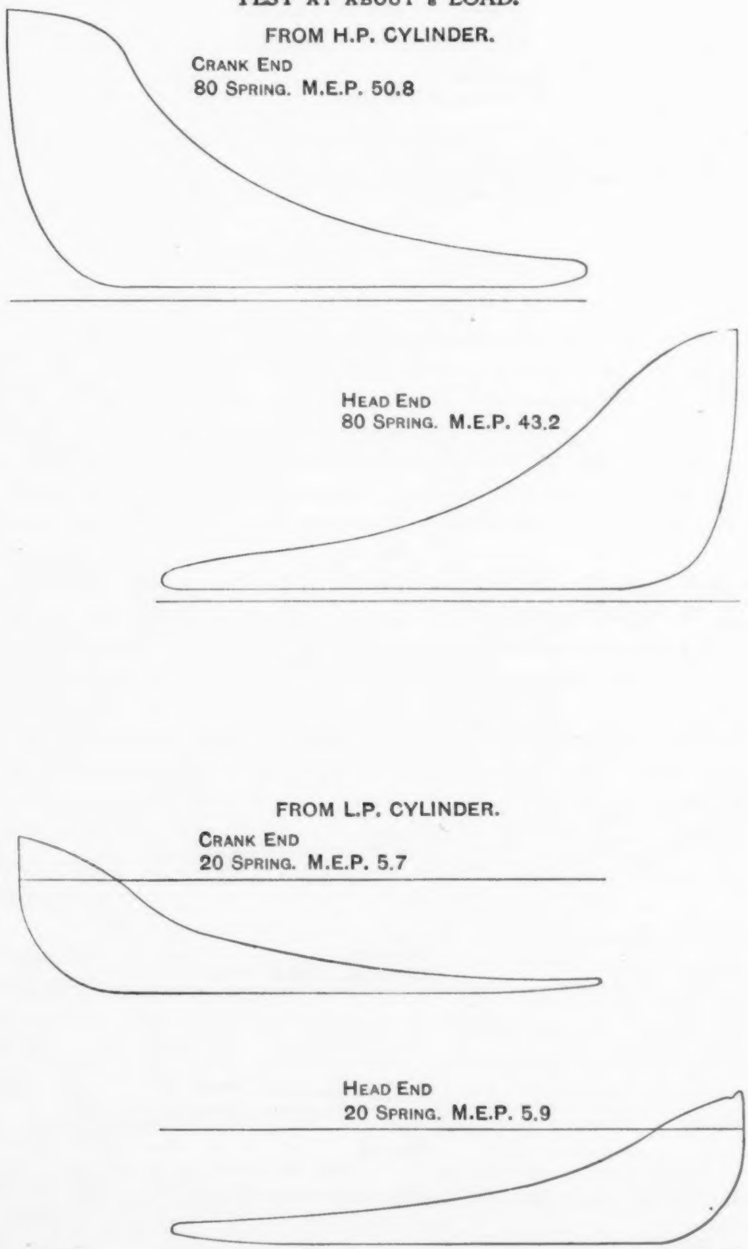
TEST AT ABOUT 1/8 LOAD.



Allen, B.T. *Am. Bank Note Co., N.Y.*

Fig. 59.

SPECIMEN CARDS.
TEST AT ABOUT ⅝ LOAD.



Allen, B.T. Am. Bank Note Co., N.Y.

FIG. 60.

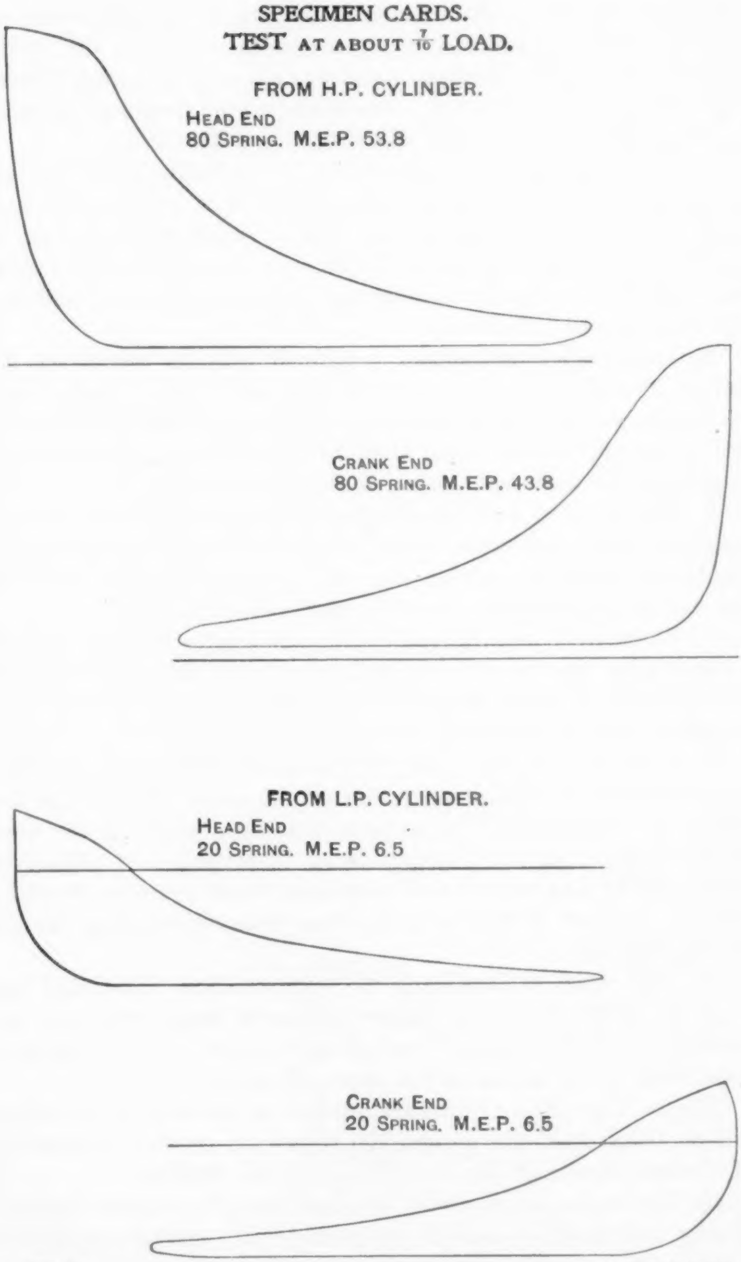

SPECIMEN CARDS.
TEST AT ABOUT $\frac{7}{10}$ LOAD.
FROM H.P. CYLINDER.
HEAD END
80 SPRING. M.E.P. 53.8
CRANK END
80 SPRING. M.E.P. 43.8
FROM L.P. CYLINDER.
HEAD END
20 SPRING. M.E.P. 6.5
CRANK END
20 SPRING. M.E.P. 6.5
Allen, B.T.
Am. Bank Note Co., N.Y.

FIG. 61.

off, such engines could be made to operate fairly satisfactory at speeds considerably in excess of that generally used on the Corliss engine; but yet the resultant speed was not as great as was desired for some purposes, especially electrical, and the design of the valve gearing still possessed many disadvantageous features.

5. After exhaustive investigations it was finally found by substituting for the detachable form of cut-off, a peculiar arrangement of bell cranks and levers, that a satisfactory amount of acceleration could be given to the valves at the points of admission and cut-off, and this by angular motion only, and unencumbered by elaborate cut-off devices.

6. This accelerated motion, combined with the advantage obtained by making the steam valves triple ported, produced a more satisfactory operation at higher speeds than was possible with the other devices, besides being absolutely noiseless in operation and requiring less care and attention for maintenance.

7. This form of valve gearing being positive in action and dependent upon a variation of the travel for the different grades of expansion, made the use of the shaft governor possible, with all its accruing advantages of speeds and regulation.

8. To secure successful and continuous service for long periods of operation the use of hardened valve gear pins and phosphor bronze boxes of ample proportion was essential, as well as the best material and workmanship obtainable.

9. It was also important, in order to insure thorough lubrication over the long periods of operation as mentioned above, that the oiling be accomplished by automatic means, which is the case in this engine, all the bearings being thoroughly and efficiently lubricated by a system of self-lubrication which requires no attention on the part of the operator, other than replenishing the oil at long intervals.

10. The result of combining the features above mentioned has been the production of an engine possessing many important advantages as the following record of the performance of a medium speed four-valve engine of this type will show.

This is, I consider of unusual interest on account of the many unique features of the design, the somewhat unusual proportion of the cylinders and the exceptional results obtained.

11. The engine was built by the Harrisburg Foundry & Machine Works, and is of the tandem compound style, directly connected to an electric generator for the purpose of furnishing power for the operation of paper mill machinery combined with electric lighting.

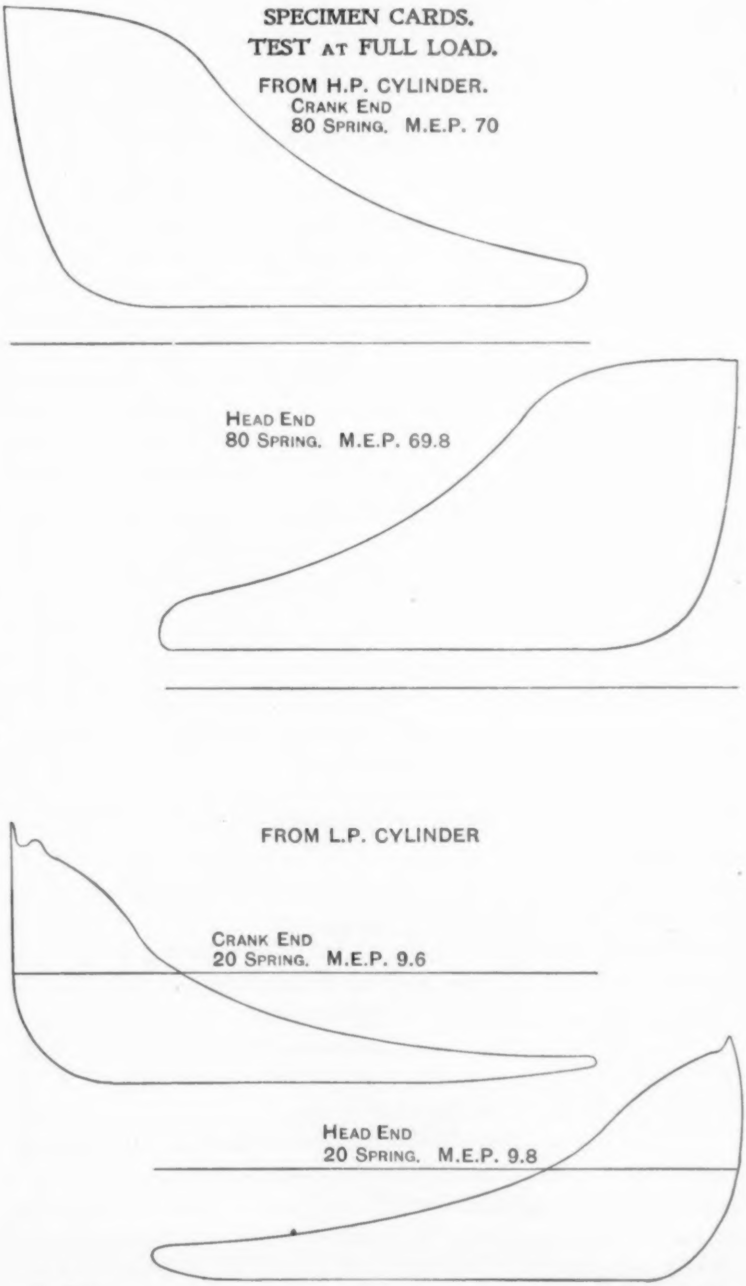
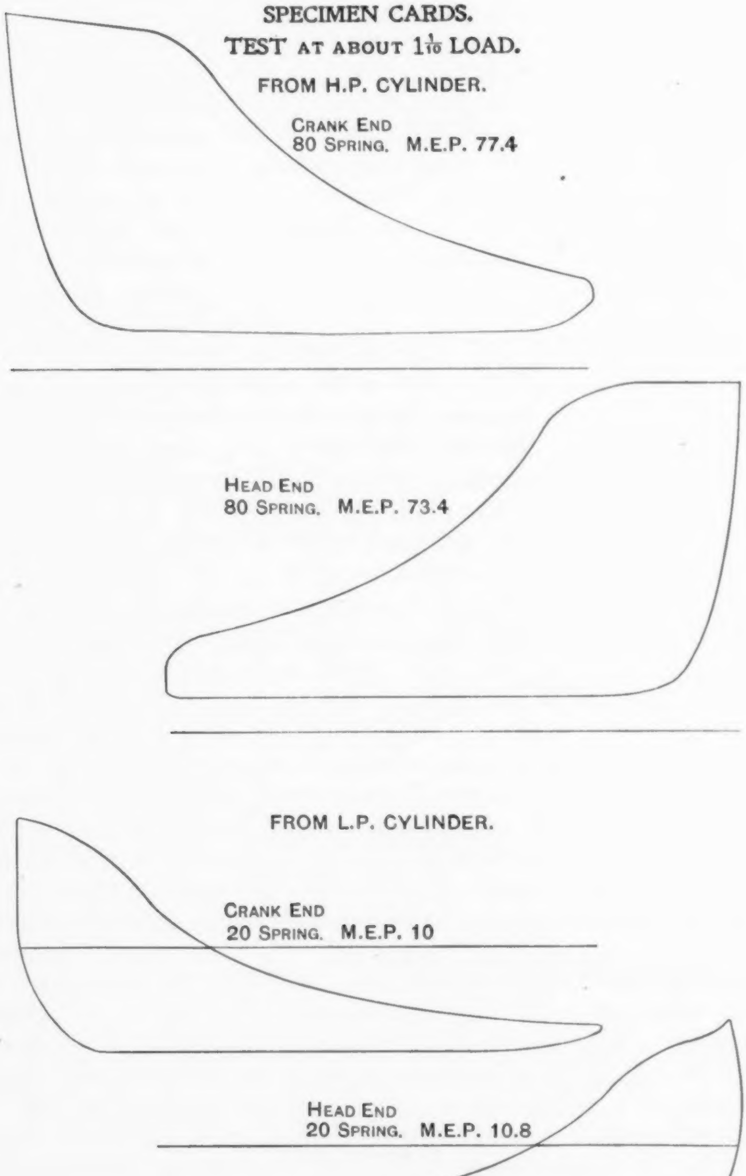

SPECIMEN CARDS.
TEST AT FULL LOAD.
FROM H.P. CYLINDER.
CRANK END
80 SPRING. M.E.P. 70
HEAD END
80 SPRING. M.E.P. 69.8
FROM L.P. CYLINDER
CRANK END
20 SPRING. M.E.P. 9.6
HEAD END
20 SPRING. M.E.P. 9.8
Allen, B.T.
Am. Bank Note Co., N.Y.

FIG. 62.

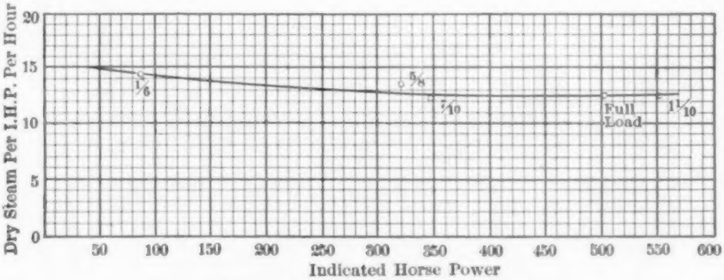


Allen, B.T. *Am. Bank Note Co., N.Y.*

FIG. 63.

Its nominal capacity is 500 horse-power at a speed of 150 revolutions per minute, 150 pounds pressure at the throttle and 26 inches vacuum. The cylinders are so proportioned as to give the high ratio of 1 to 7.33 following the style advocated by Mr. Geo. I. Rockwood, the general dimensions being given in table number one.

12. No steam jackets are used, but a vertical tubular reheating receiver is placed in the steam passage between the two cylinders, steam being admitted to the high pressure cylinder through triple ported valves of the Corliss type working in chilled iron bushings, the governing being accomplished by a centrally balanced inertia shaft or wheel governor. This governor is so constructed that it is practically balanced in all positions, being made with two inertia arms, the centers of gravity of which move in harmony with each other about the center of rotation, the balancing feature avoiding surging or violent action under all conditions of operation. This



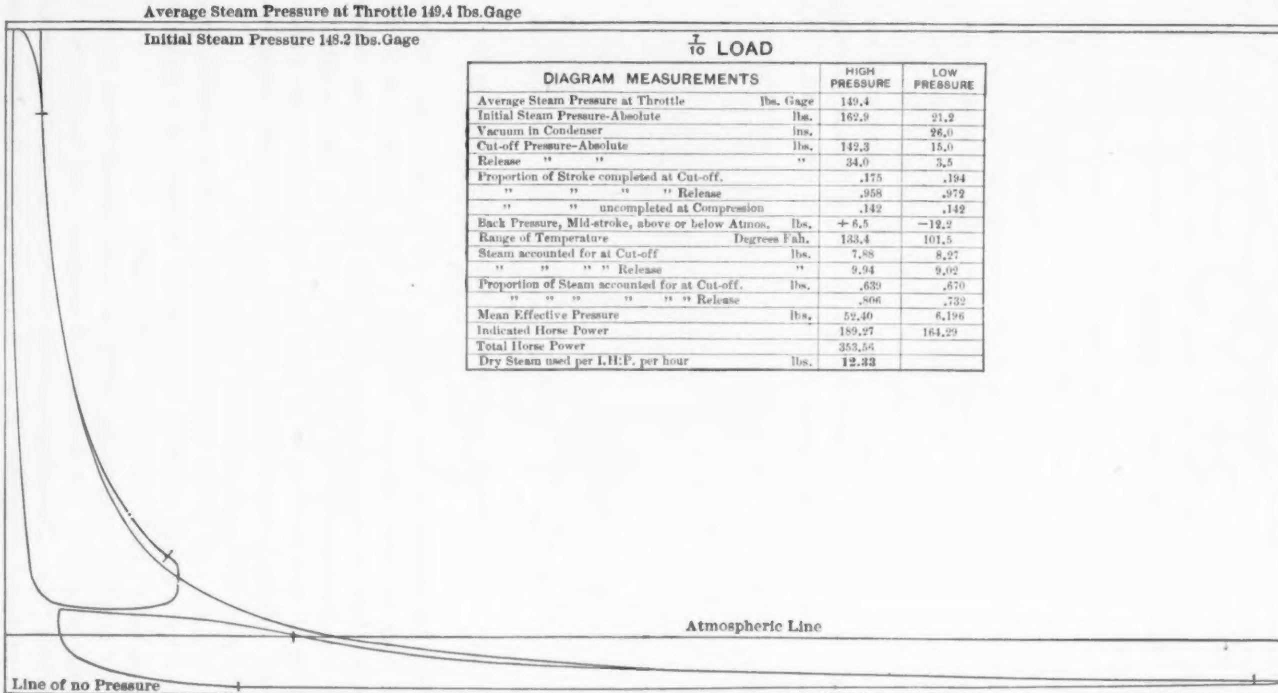
CURVE OF STEAM CONSUMPTION

FIG. 64.

operates, by means of bell cranks, the steam admission valves of the high pressure cylinder only, and is so arranged as to decrease the lead at the earliest point of cut-off. The steam valves of the low pressure cylinder are controlled by a fixed eccentric, so arranged that the cut-off in that cylinder can only be varied when the engine is not running, and remains constant, under all conditions of load and pressure.

13. The exhaust valves of both cylinders are of the Corliss type operated by a single eccentric through the medium of a peculiar arrangement of rocker arms and bell cranks.

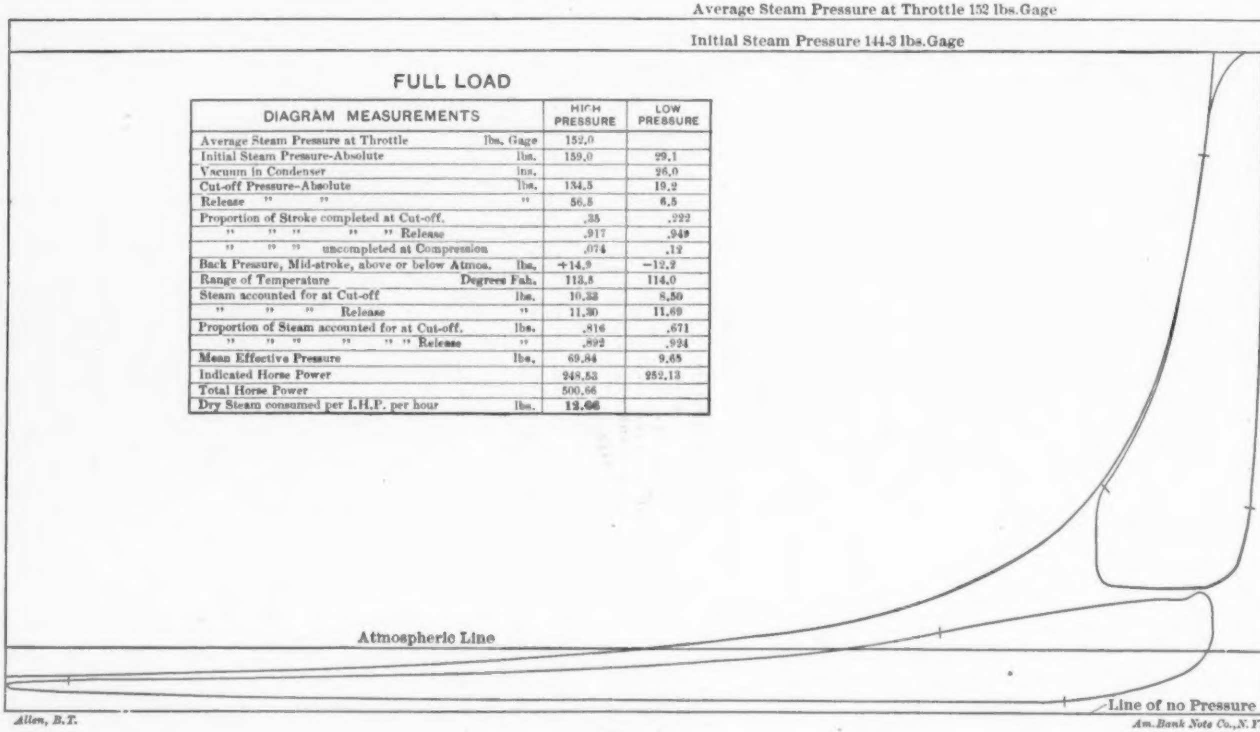
14. In making the tests the water of condensation from the exhaust was weighed at the discharge of the condenser, which was of the surface type. The steam used by the reheater was discharged from a trap and condensed in a coil and weighed, the quantities being included in the results given in the tables.



$\frac{7}{10}$ LOAD

DIAGRAM MEASUREMENTS		HIGH PRESSURE	LOW PRESSURE
Average Steam Pressure at Throttle	lbs. Gage	149.4	
Initial Steam Pressure-Absolute	lbs.	162.9	21.2
Vacuum in Condenser	ins.		26.0
Cut-off Pressure-Absolute	lbs.	142.3	15.0
Release " "	"	34.0	3.5
Proportion of Stroke completed at Cut-off.		.175	.194
" " " " Release		.958	.972
" " uncompleted at Compression		.142	.142
Back Pressure, Mid-stroke, above or below Atmos.	lbs.	+6.5	−12.2
Range of Temperature	Degrees Fah.	133.4	101.5
Steam accounted for at Cut-off	lbs.	7.88	8.27
" " " " Release	"	9.94	9.02
Proportion of Steam accounted for at Cut-off.	lbs.	.639	.670
" " " " " " Release		.806	.732
Mean Effective Pressure	lbs.	52.40	6.196
Indicated Horse Power		189.27	164.29
Total Horse Power		353.56	
Dry Steam used per I.H.P. per hour	lbs.	**12.33**	

FIG. 65.



FULL LOAD

DIAGRAM MEASUREMENTS		HIGH PRESSURE	LOW PRESSURE
Average Steam Pressure at Throttle	lbs. Gage	152,0	
Initial Steam Pressure-Absolute	lbs.	159,0	29,1
Vacuum in Condenser	ins.		26,0
Cut-off Pressure-Absolute	lbs.	134,5	19,2
Release " "	"	56,5	6,5
Proportion of Stroke completed at Cut-off.		,35	,222
" " " " " Release		,917	,949
" " " uncompleted at Compression		,074	,12
Back Pressure, Mid-stroke, above or below Atmos.	lbs.	+14,9	−12,2
Range of Temperature	Degrees Fah.	113,8	114,0
Steam accounted for at Cut-off	lbs.	10,33	8,50
" " " Release	"	11,30	11,69
Proportion of Steam accounted for at Cut-off.	lbs.	,816	,671
" " " " " " Release	"	,892	,924
Mean Effective Pressure	lbs.	69,84	9,65
Indicated Horse Power		248,53	252,13
Total Horse Power		500,66	
Dry Steam consumed per I.H.P. per hour	lbs.	12,66	

FIG. 66.

15. The steam was taken from the horizontal water tube boilers and contained from $\frac{6}{10}$ to $1\frac{1}{4}$ per cent. of moisture.

The tests were made with about 150 pounds pressure at the throttle, with the exceptions of two, one of about $\frac{5}{8}$ load with a pressure of about 130 pounds, and the other at a time when but a small portion of the mill was in operation. The load under the latter conditions being only about $\frac{1}{6}$ the rated capacity of the engine, a pressure of about 90 pounds was being carried at this time.

16. The trials were of rather short duration, but every precaution was taken to insure accuracy. The indicators, which were four in number, were attached with short pipes, one to each end of each cylinder, and operated by a positive pantograph motion with a light rod and short cord connections, the diagrams being taken at very close intervals.

17. The speeds given are the average revolutions per minute obtained from the total, registered by a recording speed counter.

TABLE I.

GENERAL DIMENSIONS OF ENGINE.

High pressure cylinder, diameter	15 inches.
Low " " "	$40\frac{1}{2}$ "
Stroke	27 "
Diameter piston rod, H.-P.	$2\frac{11}{16}$ "
" " " L.-P. { Crank end	$4\frac{3}{4}$ "
" " " L.-P. { Head "	$2\frac{11}{16}$ "
Ratio of areas of cylinders	1 to 7.33
Clearance H.-P. cylinder	3.95%
" L. P. "	4.67%
Constant for 1 lb. M. E. P. and one revolution H.-P. cylinder.	.0237
" " " " " " " L. P. "	.174

18. Table II. is a condensed report of the general results obtained under the varying conditions of load and pressure, as follows:—

TABLE II.

	Duration in hours.	Revolutions per minute.	Gauge pressure pounds at throttle.	Vacuum by gauge inches.	Horse power. High pressure cylinder.	Horse power. Low pressure cylinder.	Horse power. Total.	Steam used per I. H. P. per hour including quantity used in Reheater.
Test at about $\frac{1}{6}$ load	2	155.18	89.72	26.5	44.34	42.73	87.07	14.42
Test at about $\frac{5}{8}$ load	3	151.25	129.9	26.0	167.73	153.81	321.54	13.59
Test at about $\frac{7}{10}$ load	2	152.33	149.4	26.0	176.19	172.09	348.28	12.33
Test at about full-rated load	5	150.09	152.0	25.9	249.96	251.59	501.55	12.66
Test at about $1\frac{1}{10}$ load	2	148.89	153.0	25.5	267.24	286.25	553.49	12.73

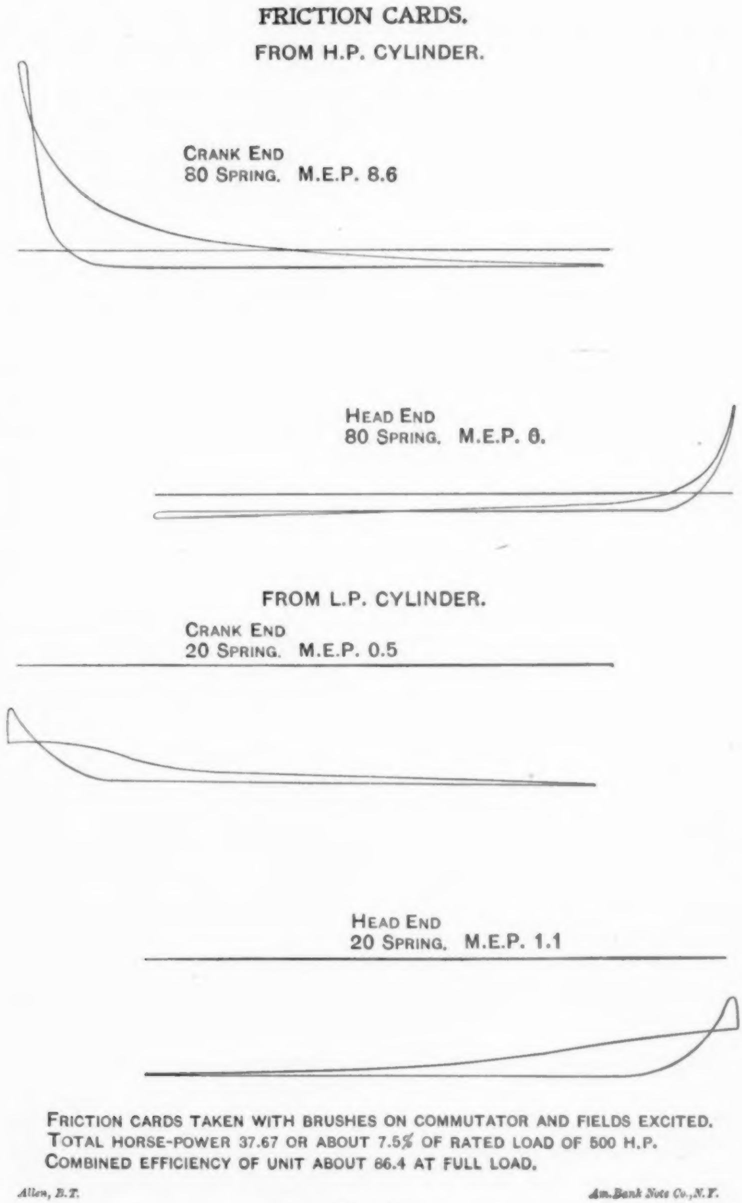
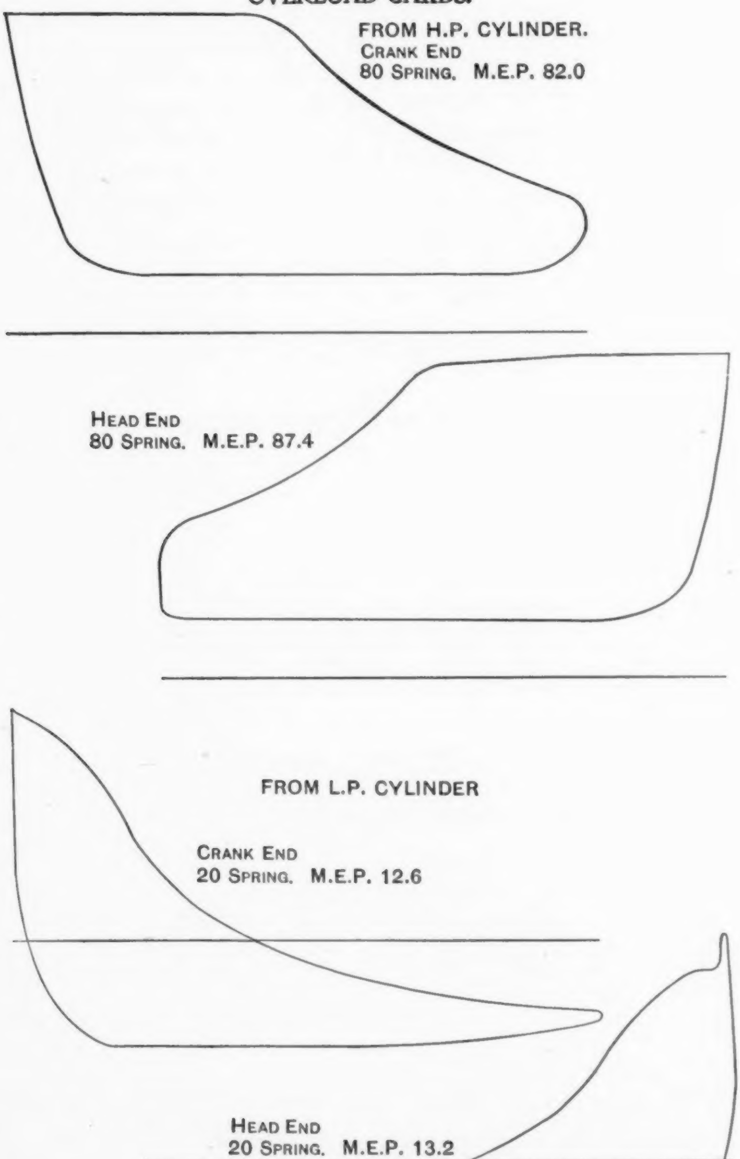

FRICTION CARDS.
FROM H.P. CYLINDER.
CRANK END
80 SPRING. M.E.P. 8.6
HEAD END
80 SPRING. M.E.P. 6.
FROM L.P. CYLINDER.
CRANK END
20 SPRING. M.E.P. 0.5
HEAD END
20 SPRING. M.E.P. 1.1
FRICTION CARDS TAKEN WITH BRUSHES ON COMMUTATOR AND FIELDS EXCITED.
TOTAL HORSE-POWER 37.67 OR ABOUT 7.5% OF RATED LOAD OF 500 H.P.
COMBINED EFFICIENCY OF UNIT ABOUT 66.4 AT FULL LOAD.
Allen, B.T.
Am.Bank Note Co.,N.Y.

FIG. 67.

OVERLOAD CARDS.



AVERAGE HORSE-POWER H.P. CYLINDER	295.09
" " " L.P. "	330.42
TOTAL HORSE-POWER	625.51

Allen, B.T. Am. Bank Note Co., N.Y.

FIG. 68.

19. Figs. 59, 60, 61, 62 and 63 are the indicator diagrams taken throughout the tests, the cards in every case being those representing the average load for the complete run. Fig. No. 64 is the diagram of efficiency or steam consumption curve.

Fig. No. 65 is of the diagrams from $\frac{7}{10}$ load combined, the cards being from the crank end of the cylinders.

Fig. No. 66 is the combination of the diagrams from the full load test, the cards being taken from the head end of the cylinders.

20. These combinations are made from the cards shown in Figs. 61 and 62, and are accompanied by tables giving the measurements taken from the diagrams. The perfection with which the cards from the $\frac{7}{10}$ load match the theoretical expansion curve, is worthy of note. This feature, however, is not so well carried out on the combined card of the full load, Fig 66.

21. I considered it of interest to add Figs. No. 67 and 68 also. The former gives the diagrams obtained from the engine when running the dynamo, without doing any work, excepting the friction of the combined unit. The latter, Fig. 68, showing a set of diagrams obtained under an over-load of about 25 per cent.

No steam consumption tests were made under these loads, the cards simply serving to show the steam distribution under these conditions.

22. A comparison of the results obtained at the different loads reveals some very interestng features; a very important one being the slight difference in the quantity of steam consumed per indicated horse-power per hour under the various conditions of load and steam pressure; representing a curve of economy closely approximating a straight line. This will be seen by reference to the diagram of efficiency. The difference between the highest and lowest steam consumption being only 2.09 pounds.

23. It will also be noticed that the distribution of work between the two cylinders is nearly uniform under all loads, up to the rated capacity, after which the tendency is for the low pressure cylinder to do the greater proportion.

24. As before stated, there is no variation of the point of cut-off in the low pressure cylinder, the setting of the valves remaining the same during the complete series of tests; and it may be that the throttling action of the governor by decreasing the lead and initial pressure in the high pressure cylinder under the light loads, contributed largely to the uniformity of the results.

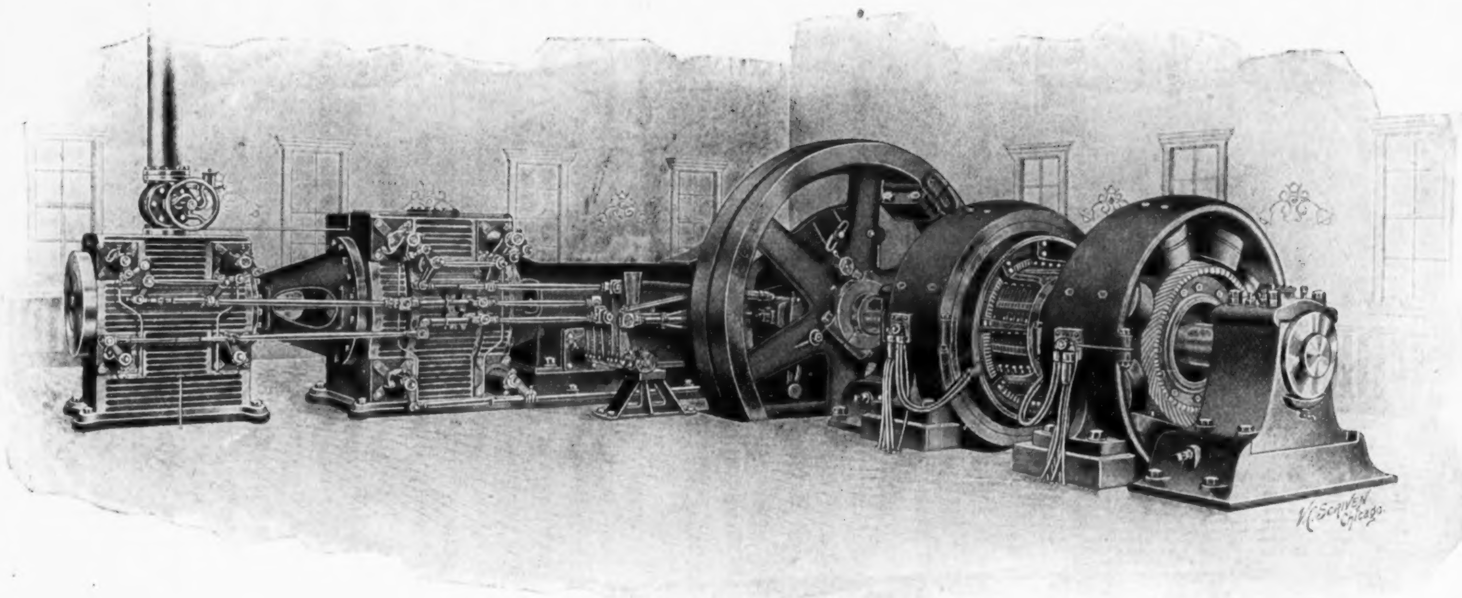


FIG. 69.

25. It will also be noticed that the best economy was obtained at about three-quarters load; this I consider due to the fact that the reheater was sufficiently large to superheat the steam passing to the low pressure cylinder up to this point. On the load being increased some accumulation of water was noticeable in the gauge glass, which would after a time evaporate and pass through the low pressure cylinder. This failure of the reheater to perform its proper functions on the increased loads, is clearly indicated in Fig. 66, and in all probability had some tendency to impair the efficiency of the apparatus under heavy loads.

26. In conclusion, therefore, it seems apparent that on account of this particular test and from others corroborating it, made at other times, the following more important and rather new principles are fairly established:

First: That as a prime mover the elaborate dash pot or other accelerated cut-off devices used in present Corliss Engine practice are unnecessary complications and unwarranted when comparing results.

Second: That the centrally balanced, direct-acting fly wheel device serves its purpose to better advantage than the indirect fly ball governor.

Third: That there is better warrant for shorter strokes and moderately high speeds than for longer strokes and resultant lower speeds, notwithstanding the element of clearances.

Fourth: That self-lubrication without additional apparatus requiring attention to secure it, enters as an improvement in net efficiency, to say nothing regarding maintenance.

Fifth: That an engine of the described design, although of marked improvement in point of simplicity will rather exceed than equal the more elaborate practice heretofore established at normal load and excels comparable prime movers in a marked degree where the work is of a widely fluctuating character.

Sixth: That considering a resulting decrease in the cost of foundations, building, floor space, and, in electric practice, generators, due to better speeds, the design described determines its importance from the standpoint of investment.

DISCUSSION.

Prof. R. C. Carpenter.—The results of the tests of the Fleming 4-valve engine described in Mr. Allen's paper show very creditable

results compared with an engine of similar dimensions operating at the relatively high speed of 150 revolutions per minute. I consider it doubtful, however, if the tests cited are sufficient to establish a water rate for ordinary cases as uniform as that shown under the peculiar conditions of the various tests cited. The tests show four results obtained when the engine carried a load more than five-eighths of its rating, and a single test when the engine was loaded to only one-sixth part of its rating; this single test, when plotted with the results obtained for higher loads, forms a nearly horizontal curve, as shown in Fig. 64 of the paper, and indicates a remarkable uniformity of steam consumption for wide variations in loading. By computing the total steam used per hour and constructing a curve with total indicated horse power as abscissæ and total weight of steam as ordinates, some interesting relations are shown which are not developed in the paper.

The total indicated horse power and total steam per hour are shown in the following table:

Total horse power.	Total steam per hour. *Lbs.*	Steam pressure. *Lbs.*
87.07	1255.5	89.72
321.54	4369.7	129.9
348.28	4294.3	149.4
501.55	6349.6	152.0
553.49	7046.9	153.0

The curve showing the relation between total steam per hour and total power developed is, as shown by the diagram, Fig. 70, which is noted, practically a straight line, *A B*, which can be represented approximately by the equation

$$\text{Total steam per hour} = 200 + 12.3 \text{ (H. P.)}$$

from which we find

$$\text{Steam per I. H. P. per hour} = \frac{200}{\text{H. P.}} + 12.3.$$

I have plotted a great many engine tests in this manner, and have found that an engine controlled by a throttling governor invariably gives a straight line curve, which fact I think was first pointed out by Mr. H. H. Willans of England, and has sometimes been characterized as Willans' Law. On the other hand, an engine controlled by an automatic governor has for its characteristic a curved line approximating that shown in dotted lines *C D E* in the figure submitted.

The governor used on the Fleming engine was of the automatic type, but the tests which were submitted were made with different steam pressures; consequently the general effect of the tests with light loads would approximate that obtained with a throttling

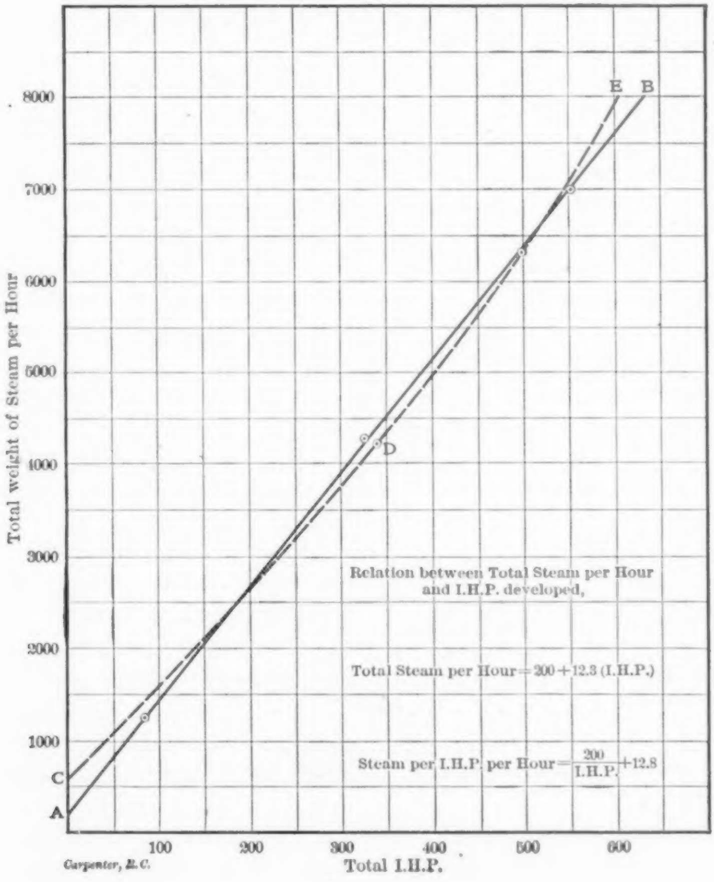


FIG. 70.

governor applied to the high-pressure cylinder, *i.e.*, a low steam pressure would give a later cut-off than would have been experienced had the pressure been high. This, in my opinion, accounts for the fact that in the curve which I submit the result of the test with the low load falls in a right line with the other tests. While this is purely accidental, it has the effect of making the steam

consumption less at a light load than would have been the case had the steam pressure been maintained at 150 pounds for the entire series of tests.

It is my impression that with a constant steam pressure, this engine will show about the same variation in steam consumption per unit of power as the Corliss engines or other good engines of the automatic type.

Mr. Rockwood.—It appears that the purpose of this paper, as stated in paragraph 1, is not quite the same thing as the impression gained by reading it. New principles of design are claimed, but none is described; and while a novel form of positively-driven Corliss valve gear is hinted at, its details are not given, and the feeling of the reader is that the author is claiming broadly, as a novelty, the use of the shaft governor as a means of controlling the cut-off valves. The same thing may be said of the references to the use of a system of automatic lubrication. Also, the "efficiency"—if by that is meant economy of steam—is mistakenly claimed to exceed that of other stationary 4-valve engines of the high ratio compound type, especially with variable loads, for the best performance of this engine is more than one pound of steam per indicated horse power per hour in excess of the best recorded performance of a high ratio compound engine of the odinary slow-speed Corliss type. I think, therefore, that the claims of paragraph 26—with the possible exceptions of the *second* and the *sixth*—should be omitted, and that the construction which it was the stated object of the paper to describe should actually be described fully, instead of merely hinted at. These strictures have reference rather to the form and claims of the paper than to the engine itself, about which enough is made clear to excite one's interest and, possibly, one's approval of its peculiarities.

The chief thing about the design of this engine which, broadly considered, is still unusual is the use of the high cylinder ratio. It is now about twelve years since the first slow speed compound stationary engine with an exaggerated cylinder ratio was built and tested. The results were published in Volume XIII. of the *Transactions.* The accuracy of these tests and their sufficiency to prove the superior value of the high ratio over the common ratio of three or four to one was not at once believed by many prominent engineers, for the reason that the general theory of the steam engine seemed to them to be against the possibility of

the truth of any such results. At that time the best acknowledged performance of the compound engine was between 14 and 15 pounds per indicated horse power per hour, and even those reports of triple-expansion pumping-engine tests, wherein was claimed a steam consumption of between 12 and 13 pounds, were at first considered to be little better than "fairy tales." If at that time an engineer had reported to this Society a paper like the one we are now considering, he would have been received with entire incredulity if not with scorn.

A distinguishing characteristic of these high ratio compounds is the phenomenon of drop. It was believed then, as it is now, that drop entails a loss of work by free expansion; and, without going further into the total effect of its use than that, there was a singular and unanimous determination on the part of all writers on the steam engine to discourage the toleration of any drop whatever in compound engines. The fact is now, however, appreciated that the high ratio compound is much more economical at light loads than is the engine with a low cylinder ratio, and the tests submitted in the paper are a further proof of this. The history of the progress of the high ratio idea is interesting, for it bears on the question of how much drop is desirable. Between 1885 and 1890 several large steamers of the Leyland Line had their old style compound engines, which used steam at 90 pounds boiler pressure, converted into high ratio compounds with steam at 150 pounds pressure. The new engines of the steamship "Algerian," for instance, were found on trial to equal the economy of triple-expansion engines of the same power. Cut-off occurred, however, at the unusually early point of one-quarter of the stroke in regular operation, with the result that these engines worked with no more drop than is usual in marine engines of the triple-expansion type cutting off at three-quarters of the stroke, as such engines generally do or even at a later cut-off.

But this system of so altering old ships received something of a setback in 1892, when a test was made on the steamship "Iveagh," both with and without the use of the intermediate cylinder. The actual figures obtained on these trials showed a saving of coal of 30 per cent. Although these tests seemed on the face of the matter to be fair comparative tests, the engines and boiler pressure being identical in both cases, yet in reality they only proved what might have been anticipated, that in the high ratio compound very excessive drop—such as where there is prac-

tically no expansion in either cylinder before release—is very wasteful. It does not do, in other words, to have all the expansion take place in the receiver. In the case of the "Iveagh" the drop was more than three times what it was on the engines of the "Algerian," being as much as 100 pounds.

On the other hand, a reasonable amount of drop is accompanied by a distinct net gain of economy at all loads. This is because the waste of heat resulting in all engines from condensation of steam and its subsequent reëvaporation at the moment of release, without the performance of work, is undeniably reduced. In the stationary 4-valve type of engine, the amount of drop in high ratio compounds is of small extent compared with what took place in the engines of the "Iveagh"; it never exceeds 30 pounds, and usually is from 15 to 25 pounds, and hence these engines do remarkably economical work, and there are in operation this minute—as the result, I believe, of the publication of the tests referred to in Volume XIII.—several hundred thousand horse power of them.

The theory upon which the engines of the "Algerian" were designed was, evidently, a wrong one. It was thought that the increased waste due to cylinder condensation, which the greater range in pressure permitted in the first cylinder, would be obviated by preventing the increased temperature range naturally accompanying it by the introduction between the two cylinders of a cylinder containing a spiral sheet of copper, and called a "heat retainer." This device was actually fitted to several steamers before the absurdity of it became apparent by actual thermometer tests. Nevertheless, had the designers and owners of those engines not fully expected that the natural increase in the range in temperature, due to the enlarged range in pressure in the high-pressure cylinder, would have been nullified by its agency, they would have abandoned the high ratio idea without experiment. My idea, on the other hand, was and is, that there would be no increase in cylinder condensation due to the omission of an intermediate cylinder, and that moreover there would be certain practical gains effected by fattening the combined diagram.

It is of interest to compare the performance of the Leavitt pumping engine at the Chestnut Hill Reservoir, Boston, with that of the high ratio Cooper Corliss compound mill engine at Providence, R. I. With identical boiler pressures, ratios of ex-

pansion and degrees of vacuums the steam consumptions per indicated horse power per hour were also identical, or 11.2 pounds.

I can corroborate the straight line shown for steam consumption at variable loads out of my own experience. I should, I confess, like to see some proof that on this short stroke engine the low-pressure clearance does not exceed 4.6 per cent. I have never seen a Corliss engine that would give less than 6 per cent. to 7 per cent. clearance even with relatively much longer strokes. Also, while the design of this engine looks, in the illustration, practical and satisfactory, I cannot see that it is any less complicated than if it had dash pots and a fly-ball governor. I believe that other things being equal except the speed of rotation—that is, with the same piston speed—the long-stroke engine will beat the short-stroke engine every time. Without doubt, however, the advantage of the long stroke is small, and out of consideration of economy of first cost of all parts of the unit, the short stroke is to be preferred.

Mr. C. V. Kerr.—I would also like to have the author give the data of that re-heater. If we had the square feet of heating surface in the re-heater, the pressure under which the steam is condensed, and the weight and temperature of the water condensed, we could make some interesting calculations on the performance of this engine that we cannot make now with the data at hand.

Mr. J. A. Seymour.—I wish to take exception to the conclusions reached in the paper concerning the advantage of a short stroke as regards economy. I have found from the results of a series of tests with engines, varying in stroke from 24 to 66 inches, but otherwise similar in all respects, that a moderately long stroke means a very considerable gain in economy as compared with a short stroke, such as that of the engine described by Mr. Allen, and that with a further increase in stroke beyond a certain point, the gain becomes much less. That as good economy cannot be obtained with engines of short stroke as with those of moderately long stroke is recognized generally by engineers having experience with both types. The remarks just made by Mr. Rockwood show that he has found this to be so.

I have mentioned this series of tests, which were made in various locations, with and without superheating receivers, some with saturated and some with superheated steam, and in each case conducted jointly by purchaser and builder to determine as to

fulfilment of contract guarantees, merely because engines of identical style and make are seldom built with such widely varying strokes and these tests, therefore, the results of which are all very consistent, afford an unusually good basis of comparing the effect which length of stroke has upon the economy of an engine.

While I agree in regard to the advantage of higher rotative speeds, I think the paper is wrong in its conclusions concerning the economy to be secured with short strokes, and that the results given need confirmation.

*Mr. Allen.**—There are some of these questions which have been asked that are pretty hard for me to make reply to at once. But first I wish to make a few remarks in corroboration of the results given in this particular test. Some two or three years ago I was called upon to make a series of tests upon a pair of engines which were of the same design exactly as this engine. This pair of engines was running at a speed of 225 revolutions per minute; the steam pressure was 125 at the throttle; the vacuum was about 26 inches, and the rated capacity was 300 horse power. It did not have any reheating receiver or any steam jackets. I think I can recollect, at least approximately, the figures obtained, and I will cite them. On the test made at one-quarter load, the results were 18.43 pounds. On the test made at one-half load, the results were 17.45 pounds. On the test made at three-quarter load, the results were 14.97 pounds. On the test made at about the full rated load of 300 horse power, the result was 15.22 pounds, and on the 25 per cent. over load 15.33 pounds. These figures represent the actual weight of feed water pumped into the boilers and without any corrections for moisture. If you will compare the figures given, the highest and lowest, you will notice that in this case also the difference is only about 3.46 pounds. The ratio of this engine was 1 to $5\frac{1}{3}$. It had a high-pressure cylinder of 13 inches. Low pressure thirty and five thirty-secondths, and the stroke was $17\frac{1}{2}$ inches, running at a speed of 225 revolutions per minute. I have made other tests of this particular type of engine which corroborate the tests made in this paper.

With reference to the remarks of Professor Carpenter regarding the variation in steam pressure, I wish to say that the governor is so arranged on the shaft of this engine that by giving it a certain amount of angular advance, the lead, as the paper states,

* Author's closure, under the Rules.

is reduced at the earlier points of the cut-off. If you will measure the initial pressure given on the card of the sixth load in this paper, you will notice that it is far from being 90 pounds in the cylinder, and that the throttling action is very apparent.

It has been my experience that this same engine running with a higher pressure does not give any higher initial pressure in the cylinder on the light loads than it would if the pressure at the throttle were low. So that, outside of the fact that there is a little more heat in the steam chest, I do not believe that the higher pressure would have made any difference in the result.

Mr. Kent asked for further data regarding the design of the engine. I will add that to the paper before it is finally published.

Regarding the clearance, the valves of this engine are of rather peculiar construction, and I really do not think that the engine can be compared with a Corliss engine—that is, not in the same sense that the discussion seems to imply. The clearance between the low-pressure piston and the head was only about one-eighth of an inch. The head was turned and polished, as also was the piston, on both sides, the crank end being cast in solid. The percentage of clearance given is that percentage which has been carefully calculated from the drawing and checked up by two or three different persons.

Regarding the condenser, this was of the surface type—I cannot remember the name of it. It was operated independently of the engine. There was no test made of the amount of steam consumed by the condenser.

With reference to the reheater, steam direct from the boiler was passed through the reheater and carried off by means of a trap. I am sorry to say that through an error on the part of the man looking after the water coming from this reheater, he neglected to take the temperature of it. So I cannot give you that data, but the weights I am absolutely certain are correct; that is, the weghts of water consumed by the reheater. These I have in the original tests, but not in this paper. That data of course I could put in the final printed matter also.

In response to numerous requests for more complete details of the construction and further information of the test of the 4-valve engine, which was the subject of my paper, I have considered it advisable to add the drawing Fig. 71, showing cross section of valves and cylinders, and Fig. 72 showing the plan and elevation of the engine in detail.

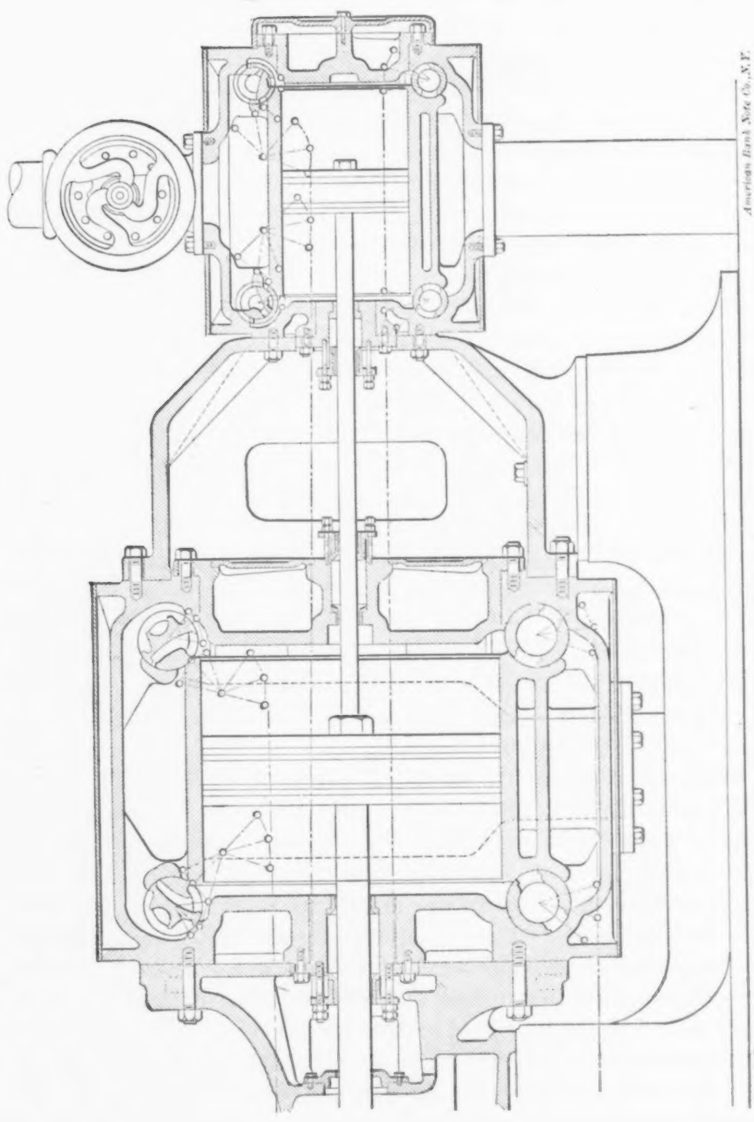


FIG 71

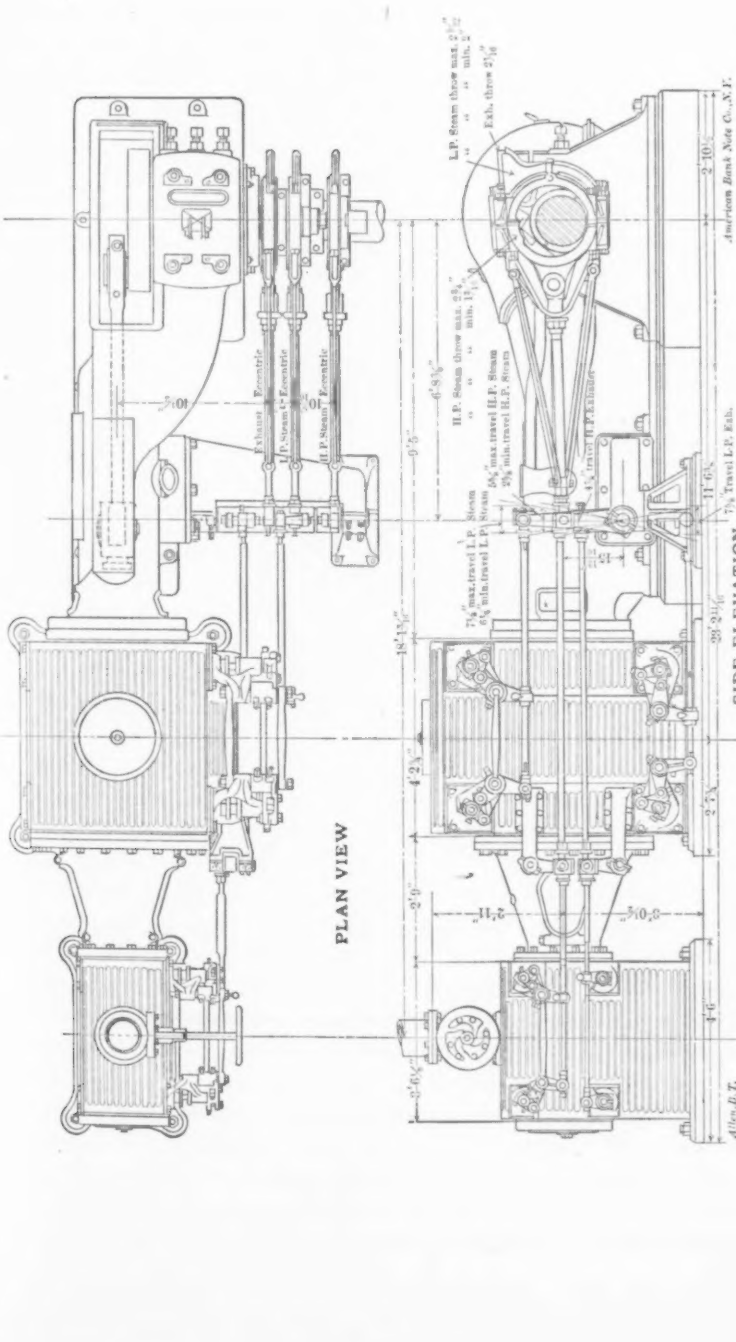


FIG. 72.

The drawing of the cylinders, Fig. 71, was made for a smaller engine, but the design and construction of the valves is the same as on the engine from which the test was made.

It will be noticed on referring to this cross sectional drawing,

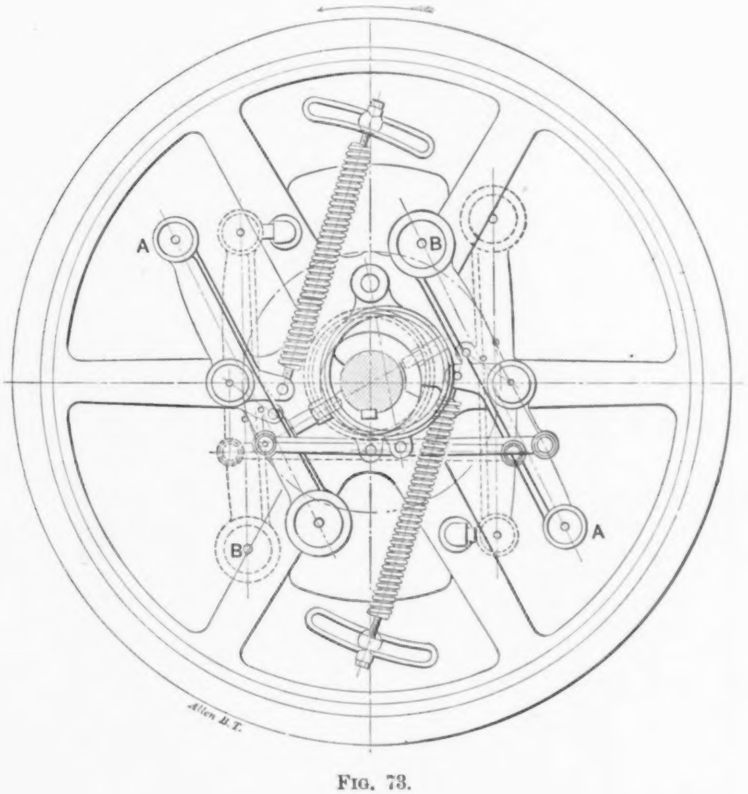


FIG. 73.

that the steam valves of the high-pressure cylinders operate in removable cast-iron bushings, as described in the paper, and that the ports are arranged in such a manner that the three edges are opened at one time. This arrangement of triple ports combined with the rapid angular motion, obtained from the use of the peculiar arrangement of bell cranks, gives a very quick admission and cut-off. The use of the bushings gives a ready means of renewal when repairs are necessary, as the cages can simply be

removed and new ones forced in, in a very short time, obviating the necessity of boring out the holes as must be done in the case of the Corliss engine.

The steam valves of the low-pressure cylinder are also triple ported, but have a different arrangement of steam passage and do not operate in bushings. We do not consider this necessary in this cylinder, as the valves are constantly working at the same travel, insuring thorough lubrication, and as the surfaces are large, the amount of wear is very slight after years of operation.

The exhaust valves in both cylinders are single ported, the valves being virtually plug cocks which close in such a manner as to eliminate entirely the space inside the valve from the clearance volume. The valves of the high-pressure cylinders are operated by means of our centrally balanced inertia governor, which is clearly shown by Fig. 73. This governor is the same as is used on all engines of the Fleming system.

The steam valves of the low-pressure cylinder are operated by means of an independent eccentric, which is practically fixed to the shaft, but has a point of suspension similar to that used on the governor, and is arranged with a screw by means of which it can be moved across the shaft, and the point of cut-off varied in the low-pressure cylinder without changing the lead. This can be done only while the engine is not in motion.

The eccentric next to the main bearing controls the exhaust valves in both high and low-pressure cylinder, by means of a peculiar combination of rocker arms. These rocker arms are all made of open hearth steel castings, and those which operate the steam valves are keyed to the shaft at each end next to the rocker shaft bearings.

This shaft has a running fit in two bearings attached to a bracket which is bolted to the side of the bed. Between these two rocker arms at the extreme end of the shaft is located the rocker arm which connects to the steam valves of the low-pressure cylinder, also the rocker arm which operates the exhaust valves of both cylinders. These rocker arms run loose on the shaft and are babbitted inside the hole to avoid cutting. They are also provided with large bronze adjusting shoes to take up the wear at this point.

All the pins used throughout the valve gearing, and in fact on the whole engine, are made of steel, case hardened and ground, and the connections in the valve gearing are of phosphor bronze,

provided with means of adjustment so that wear can be taken up without removing the rods from their pins. All the bell cranks and valve arms are made amply strong of malleable iron to guard against possible breakage.

There is a disconnecting valve for each pair of valves where the reach rod connects to each set of bell cranks. This disconnecting device is arranged in such a manner that by simply throwing a lever, a cam is disengaged from the rod, which is then free to move through the pin connected to the bell crank, so that by means of a starting bar each pair of valves can be readily tried before the engine is turned over under steam.

The valve gearing is most substantial throughout, all the parts being exceptionally large and heavy. The brackets of the steam valves are connected together with links and ream-fitted bolts in such a manner that the two steam brackets of each cylinder form a rigid truss and are made to distribute the strain in an admirable manner.

The same system of self lubrication is used on these engines as is used on all the Fleming engines. The cylinders are neatly covered with cast iron lagging, ground and polished. The space between being filled with the best quality of non-conducting material. The connection between the high- and low-pressure cylinder is arranged in halves, and attached to the cylinders by means of cap bolts, so that in case it is necessary to examine the interior of the low-pressure cylinder the connection can be readily removed, thus allowing ample room to move back the head and examine the cylinder or make adjustments to the piston.

The piston of the low-pressure cylinder has a bull ring of the phosphor bronze type, which is arranged in such a manner that in case of wear the piston rod can easily be raised. The use of phosphor bronze in the bull ring has proven very satisfactory and insures long wear without cutting.

The drawing of the plan and elevation of the engine, Fig. 72, shows more clearly the style and construction of the valve gearing than does the cut, Fig. 69, in the paper read before the Society, which cut was made from a photograph of the engine.

In order to show more clearly the arrangement of the re-heating receiver and its connections on this particular engine, I have considered it of interest to add drawing, Fig. 74, which is a foundation plan of this particular engine.

It will be noticed that the re-heating receiver is of the vertical

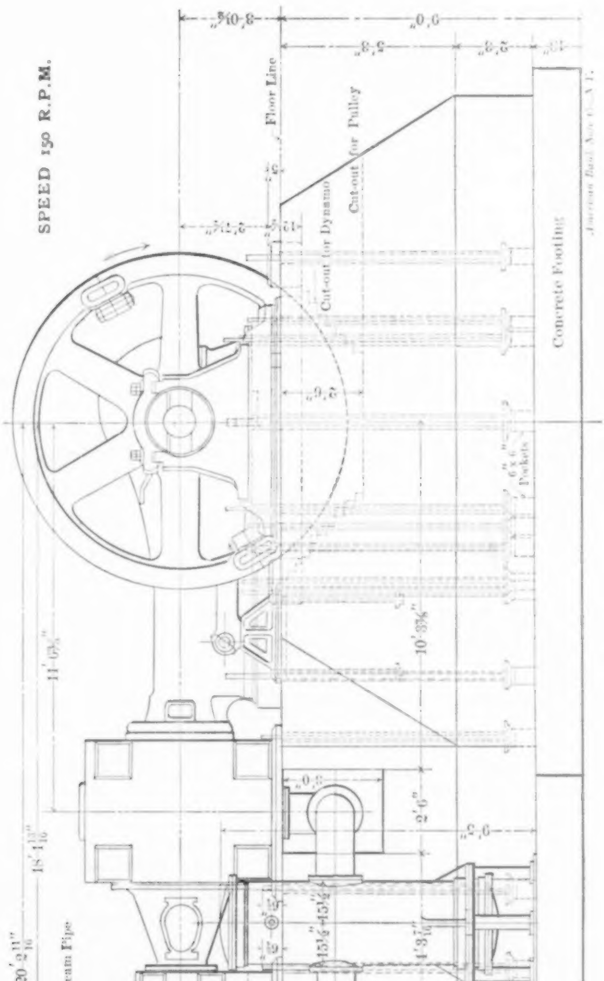

SPEED 150 R.P.M.
Floor Line
Cut-out for Dynamo
Cut-out for Pulley
Concrete Footing
Steam Pipe
Pockets

Fig. 74.

type. It was built by the Wheeler Surface Condenser Co., and contains a series of vertical tubes, one inside the other, arranged after the manner usual in the construction of their surface condensers.

This re-heating receiver contains 250 square feet of heating surface, and the writer is of the opinion that the falling off of the economy on the full load is largely due to the fact that this surface was insufficient for this size engine. The amount represents about $\frac{1}{2}$ square foot to the horse power at the rated capacity of the engine.

In reply to question asked by Mr. C. V. Kerr with reference to the re-heater. The steam entered the re-heater at 150 pounds pressure, and was taken therefrom by means of a Bundy trap and passed through a coil of pipe immersed in a tank which was constantly running full of cold water. This tank was placed above another tank on a pair of scales, so that when the trap discharged its contents, all the steam passing through the re-heater was allowed to flow into the tank on the scales where it was weighed.

As mentioned by the writer, there was no temperature taken of this water, but it is fair to say that it was not above 100 degrees.

The amount of water used by the re-heater under the different loads given in the table of the test No. 2 of my paper is as follows:

Test	at	$\frac{1}{6}$	load,	steam	used	per	I. H. P.	per	hour	1.26	lbs.
"	"	$\frac{5}{8}$	"	"	"	"	"	"	"	1.21	"
"	"	$\frac{7}{10}$	"	"	"	"	"	"	"	.62	"
"	"	full	"	"	"	"	"	"	"	.54	"
"	"	$1\frac{1}{10}$	"	"	"	"	"	"	"	.54	"

Referring to remarks made during the discussion in which exception was taken to the conclusions reached in the paper concerning the advantage of the short stroke engine as regards economy, the writer would say that he has been called upon to make tests of a great number of engines similar to the one described in the paper, and it is fair to say that, in each and every case, he has found the economy to exceed that of the Corliss engine, either medium or long stroke, under similar conditions.

The results cited by the writer on the engine of 300 horse-power, with a stroke of $17\frac{1}{2}$ inches, speed 225 revolutions per minute, were at most economical points less than 15 pounds, with 125 pounds of steam and 25 inches vacuum and without any re-heating receiver.

The writer has also in mind another test which he was called upon to make on a simple four valve engine of this type, size 15 × 15; speed, 180 revolutions; horse-power, 135; steam pressure, 120 pounds at the throttle, non-condensing. The results obtained from this engine were 23.4 pounds per indicated horse power per hour, which is as good as the best obtained from either long or medium stroke Corliss machine or other engines of similar design.

Referring to Professor Carpenter's part of the discussion: the writer's experience has been that, with the governor advanced as is done on the Fleming four valve engine (thus decreasing the lead to zero at the lighter loads), in controlling the speed of the engine, the initial pressure must be throttled as the steam enters the high-pressure cylinder, especially on the light loads; if this were not the case the engine would run away, therefore the lighter the load the less the initial pressure in the cylinder. This is true no matter what the pressure is at the throttle, so had all the tests been made with the same pressure at the throttle, in the writer's opinion the results could not have varied greatly. The engine carries a certain load at a certain speed, and as the regulation is very close for all changes of load, the initial pressure is cut down in order to maintain a speed practically constant under varying loads.

Referring to Mr. Rockwood's remarks during the discussion, the clearance in the low-pressure cylinder of this engine was determined, as mentioned by the writer, by carefully calculating the volumes from drawings.

The pistons ran within about $\frac{1}{8}$ inch of the cylinder head. The work of calculating this clearance was very carefully done and was gone over by several different people, so that the writer is confident that the calculations are correct.

A reference to the cross sectional drawing, Fig. 71, will show that the valves lay very close to the cylinder, and that there is no clearance in the exhaust valves at all, they being single ported, and the only portion of the steam valves which entered the clearance is the narrow part through the valve.

It will be remembered in the ordinary form of Corliss exhaust valve there is quite an amount of space in the valve which must be taken into the clearance, this possibly has something to do with the comparatively small amount of clearance in the cylinder of this type and stroke as compared with what can usually be obtained with a Corliss form of valve.

The claims made in the paper regarding the economy of the engine are intended to mean that the net efficiency of an engine of this type is higher than can be obtained from the long-stroke, slow-speed engine, because of the combination of low steam consumption per horse power; more perfect regulation on account of the use of the shaft governor; economy of oil on account of self lubrication; reduced amount of floor space and foundation as compared with the slow speed engine, and if direct connected very much reduced cost of generator on account of the high rotative speed.

No. 1023.*

A COMPACT GAS ENGINE: BEAM TYPE.

BY C. H. MORGAN, WORCESTER, MASS.
(Member of the Society.)

1. Having had an experience of 35 years in the manufacture and use of gas-producers and producer-gas, it has been the pleasure of the writer to feel something of the possible future of the gas engine and to endeavor to add something to its development and perfection.

2. This has been done collaterally to other and more pressing demands but always with unabated confidence and interest.

3. In common with others the enormous gaseous wastes of the blast furnace and coke ovens has been a tempting field for research and exploitation. Our esteemed fellow member, Mr. H. H. Campbell, in his "Manufacture and Properties of Iron and Steel," strikingly calls attention to the fact that while the blast furnace is primarily a producer of iron, it is a gigantic gas producer as well.†

4. Like the dependence of the steam engine upon its boiler for uniformity and quality of steam so is that of the gas engine upon the producer for suitable gas. Nor is such gas easily obtained. Blast furnace and producer gas, especially the former, contain large amounts of dust which comes from the ores, limestone and ash, and which is driven out with the blast. This dust has been the *bete noir* of the gas engineer in attempted utilization of the gas. Clogging the checker work and flues when used for reheating, impeding combustion when burned under boilers, and cutting and

---

* Presented at the New York meeting (December, 1903) of the American Society of Mechanical Engineers, and forming part of Volume XXV. of the *Transactions*.

† Mr. Campbell has made a valuable suggestion in his "Metallurgy of Iron and Steel." (See page 124.) He says: "It is quite possible that the exhaust gases from the gas engine can be profitably employed to heat the blast in the stoves of the blast furnace."

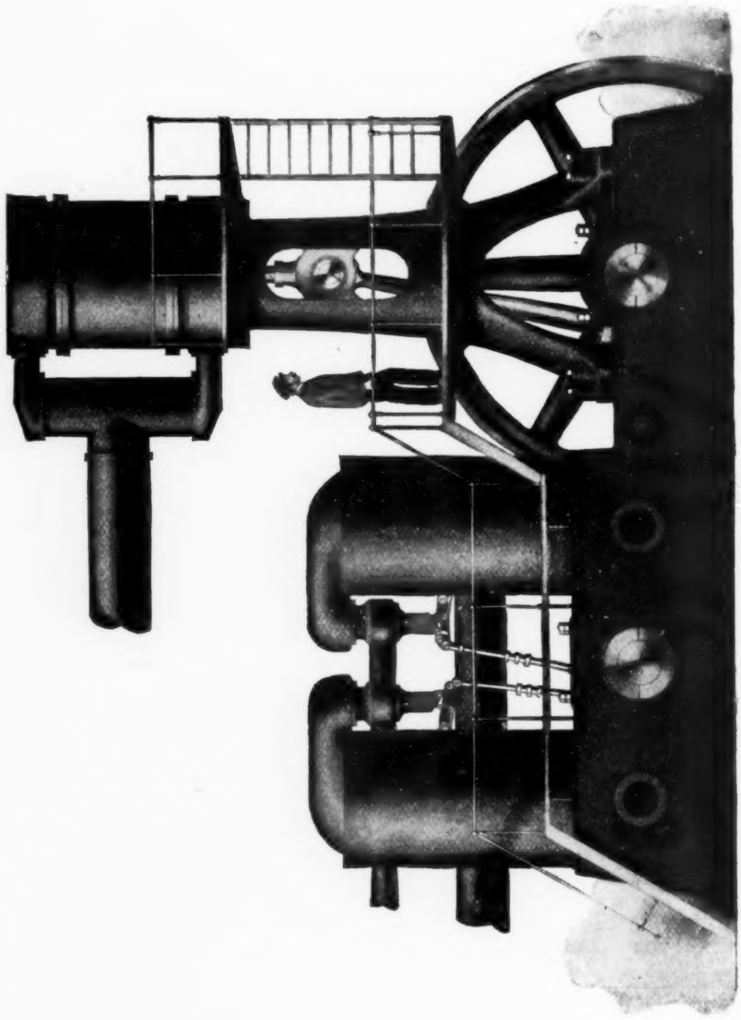


Fig. 75.

grinding vital parts of the machine when used in gas engines. Various devices for "washing" or eliminating the dust have been devised, when, in the year 1900, German engineers discov-

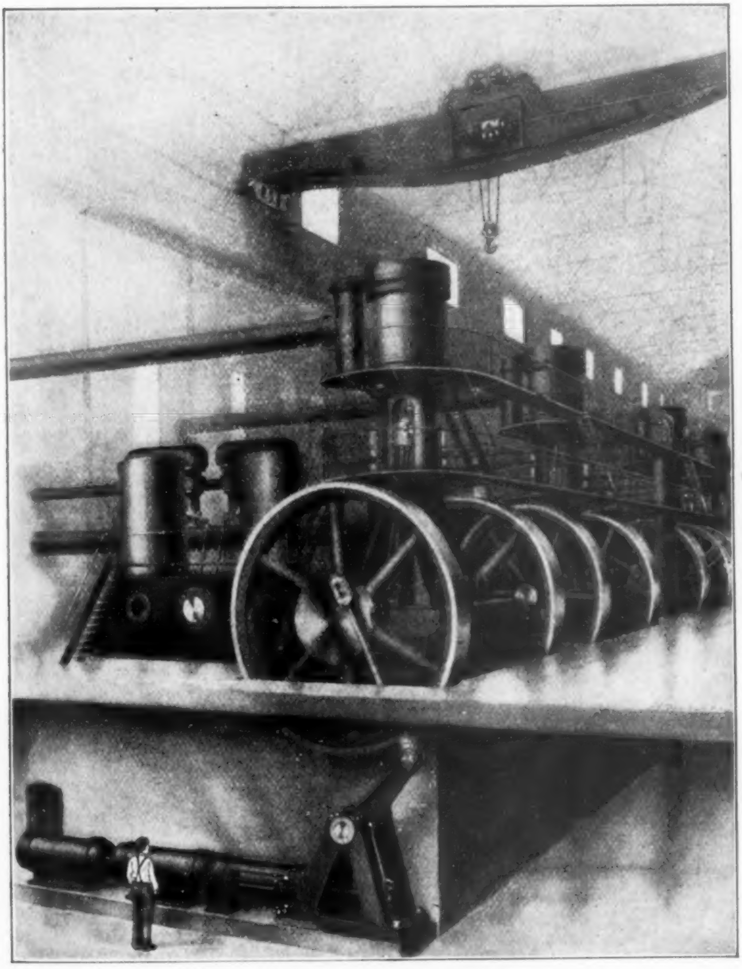


FIG. 76.

ered the value of passing the gas through a simple centrifugal fan blower injected with a small spray of water. The result has been exceedingly promising and when perfected will doubtless solve the troublesome problem.

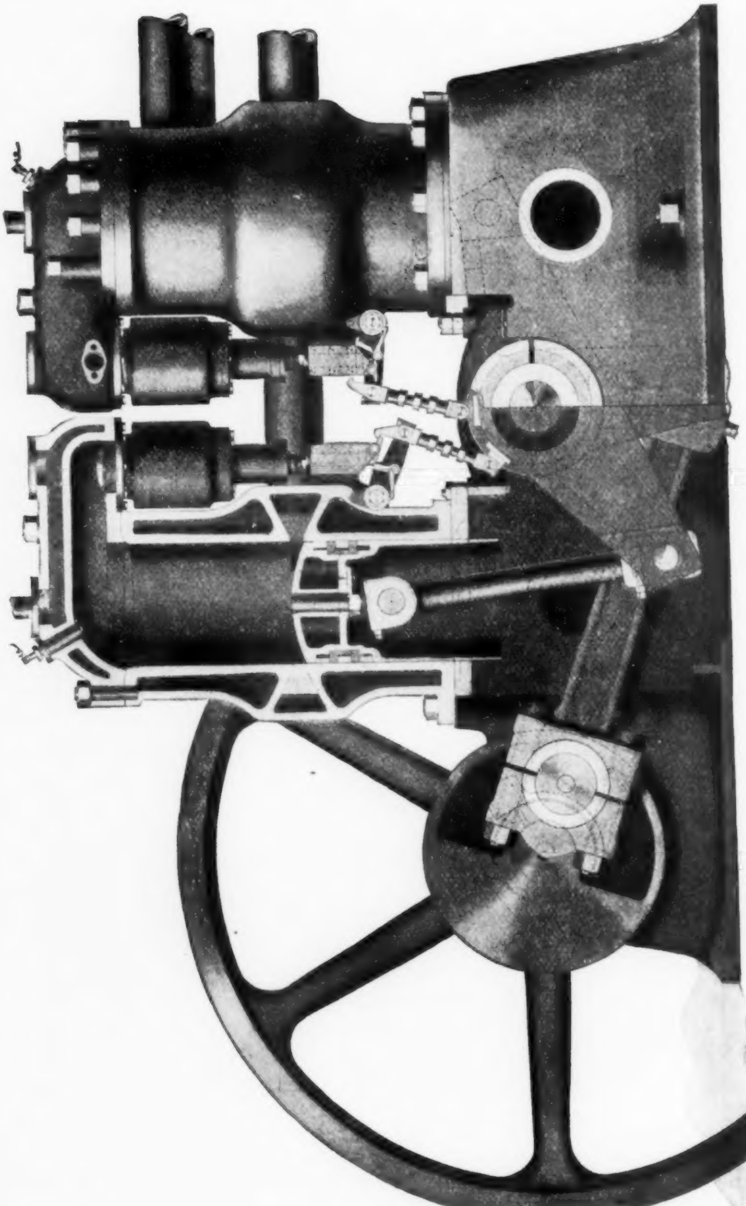


FIG. 77.

5. This engine, as illustrated in this paper, operates on the two-cycle principle with an explosion at every downward stroke of the piston. Two compressors, one for gas and one for air, furnish the engine with its charges of gas and air, and provide air for seavenging. The exhaust is through ports in the cylinder which are uncovered by the piston near the end of its downward

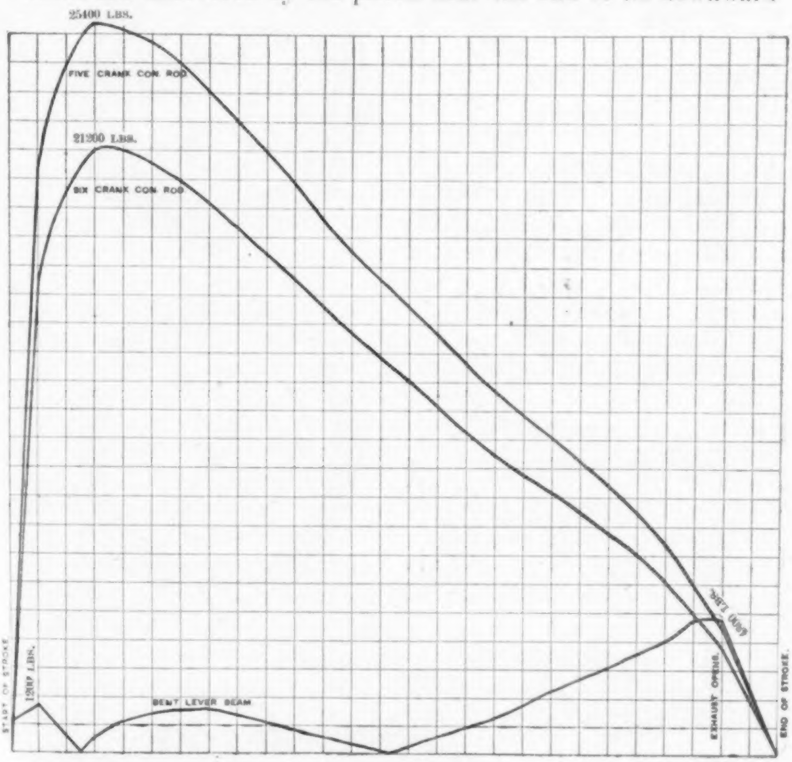


Fig. 78.

stroke; thus avoiding the use of water-cooled exhaust valves and the mechanism which would be required to operate them. It can also be operated on the four-cycle system by slight changes.

6. The engine herein described has been designed with special reference to the use of such blast furnace and producer gases. the working beam type of engine and bent lever beam have been chosen for the following reasons:—

7. It allows the cylinders to be placed upright, the balancing of the pistons and their connecting rods; it subjects the cylinder

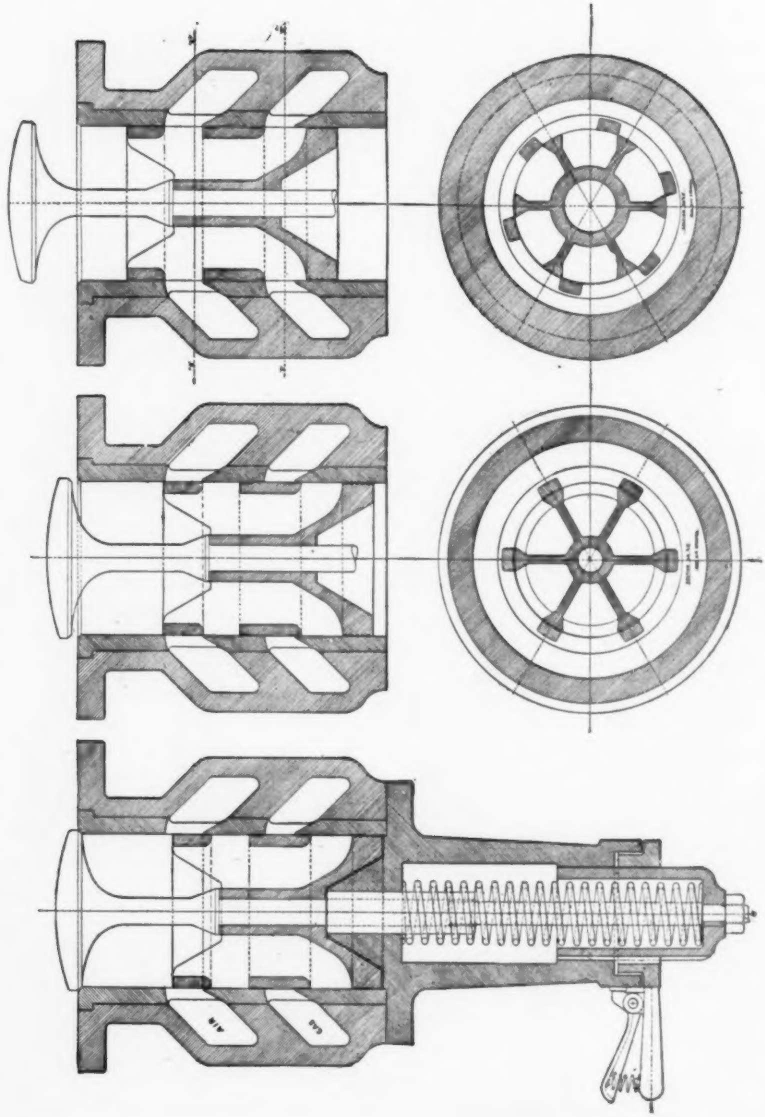


FIG. 79.

walls to much less pressure from the pistons, reducing the friction to a minimum. During the first half of the working stroke the average pressure of the piston on the walls of the cylinder is 100 pounds per square foot of surface. Figure 78 shows diagrammatically a striking comparison between the beam type of engine and the ordinary horizontal type with trunk piston direct connected to crank shaft. Two cases of the latter type are illustrated, one with the connecting rod of a length of 5 and the other 6 cranks respectively. The abscissæ represent points in the working stroke while the ordinates represent total pressures in pounds on the cylinder wall due to the angularity of the connecting rods. The distance between each of the horizontal cross lines is equivalent to 1,000 pounds, and the distance between the vertical cross lines to two inches of piston stroke.

8. The bent lever beam also gives us the least pressure against the walls of the cylinder during that part of the working stroke when there is the highest speed of the piston, and also facilitates the lubrication of the piston and the piston connecting rod pins.

Compactness.

9. The John Cockerill Co., of Seraing, Belgium, who were pioneers in the construction of large gas engines using blast furnace gas, purpose exhibiting a 3,000 horse-power gas engine at the St. Louis Exposition in 1904. An official statement gives the floor space to be occupied by this engine as 85 feet x 45 feet (3,825 square feet). A pair of engines of the beam type arranged for blast furnace duty with a capacity of 3,000 brake horse-power, although of the same power as the Cockerill engine require a space of only 24 feet x 32 feet (768 square feet) one-fifth the space.

10. Each of the inlet valves of the engine is a combination of the mushroom or poppet valve and cylindrical valves with ports for gas and air, all the valves being arranged on one stem. The cylindrical valves have radial partitions which facilitate the mixture of gas and air before entering the cylinder, thus securing prompt ignition at the first part of the working stroke and a corresponding high efficiency. The air ports and poppet valves are opened in advance of the gas ports so as to thoroughly scavenge the cylinder. Figure 79 shows the inlet valve in section in three characteristic positions. The left hand figure represents the valve closed, the central figure in a scavenging

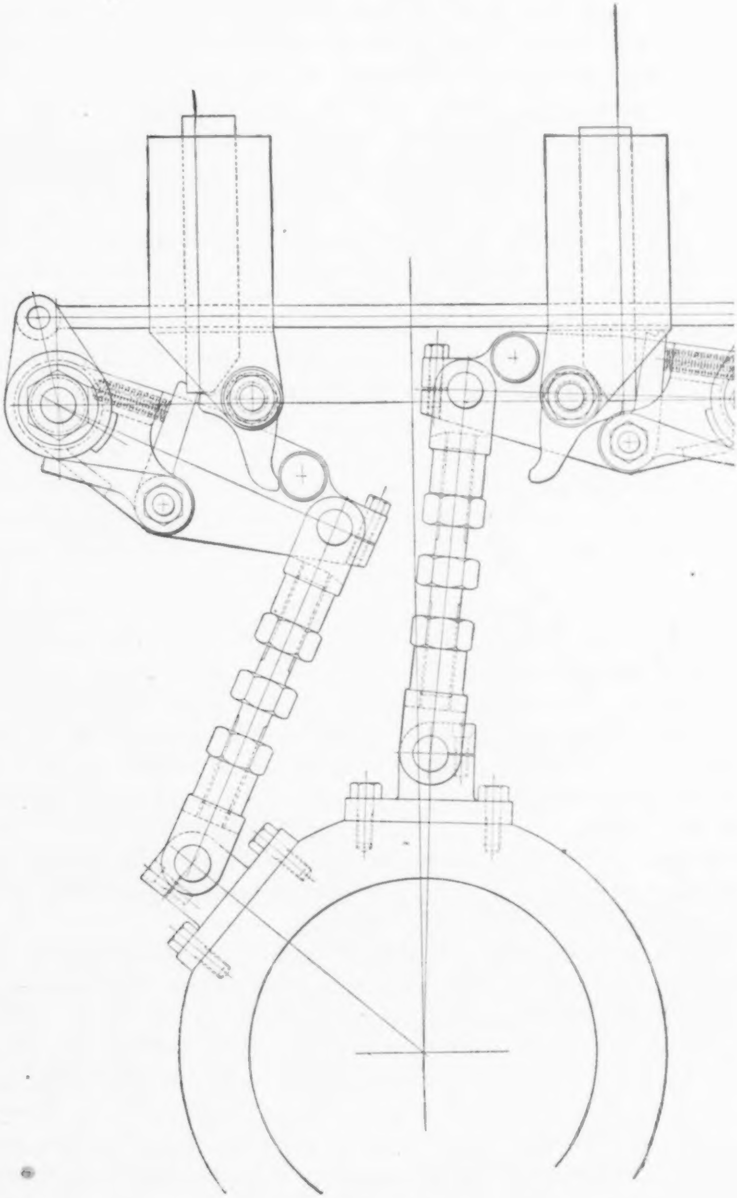


Fig. 80.

position and the right hand figure the position of the valve when fully open. The proportion of gas and air can be varied while the engine is running by rotating the valve slightly upon its axis by means of the handle shown in the illustration.

11. The valve gear is of the releasing or Corliss type, with dash pots to secure quiet closing. The valve gear is operated directly from the working beam without eccentrics or cams, which reduces the lubrication and care usually required in steam and gas engines. Figure 80 shows the valve gear as adapted for use in a two-cylinder engine. The piston of the left hand cylinder has completed its downward stroke, the scavenge lever during the latter part of this stroke has raised the valve to the position shown in the illustration, which allows the trigger to engage with the valve slide and continue the raising of the valve, opening the gas ports, as the piston starts on its upward stroke.

12. This trigger is under direct control of the governor which effects through its connections the tripping of this trigger at varying points in the stroke, as determined by the load on the engine. This tripping allows the valve to return to its seat, the dash pot preventing any shock due to closing. The right hand valve gear is shown in its highest position, the trigger having been tripped by the action of the governor.

13. Each cylinder of the engine is provided with two ignition or spark plugs so that in case of failure the current can be switched from one to the other without stopping the engine. The timing of the ignition, so that the spark may come before passing the center, at the center, or at any required part of the working stroke is accomplished by a movable disc which can be changed instantly to any required position when the engine is running at full speed.

14. Fig. 75 shows a two-cylinder engine of the beam type with two vertical single acting motor cylinders and one vertical double acting blowing cylinder of 1,500 brake horse-power. The connecting rods work on the same crank pin and are at an angle of 90 degrees, an arrangement of recognized merit.

15. Fig. 76 represents the interior of a power station with overhead travelling crane. The group of engines shown is made up of units of 1,500 horse-power each, using blast furnace gas and arranged for furnishing blast for the furnaces.

16. Fig. 77 shows an elevation partly in section of the engine without blowing tube.

17. Fig. 81 is a connection diagram for a beam and a direct connected type of engine. It shows the amount of angularity of the connecting rods and the points in the stroke at which the angularity is a maximum.

18. These then are the lines along which my efforts have been

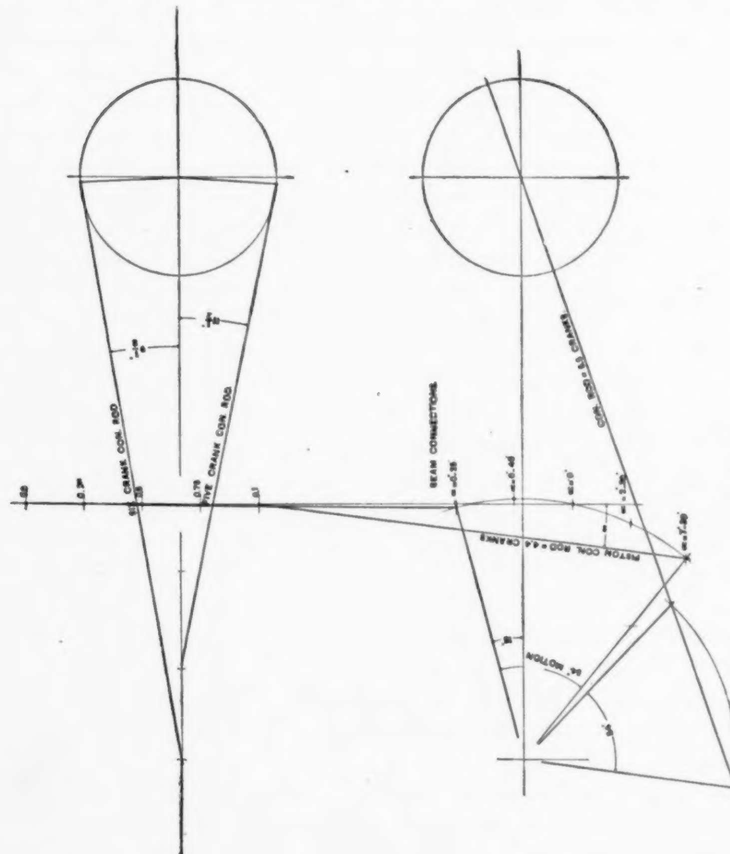


FIG. 81.

made. They are brought to the attention of the Society with the hope that they will give some measure of interest and enthusiasm in this highly important field.

19. If the wastes of the old generation are the profits of the new then large dividends are waiting us in the squandering of the past.

20. The fearful wastes of blast and coke furnaces; the enormous power in them awaiting capture and control must impress any one at all sensitive to manufacturing economies.

21. May this new motor assist in bringing quick subjection, harness a new and better power for the wants of man, and so obey the divine command "to subdue and have dominion."

22. In closing, the author desires to express his thanks to his assistant, Mr. A. J. Gifford.

DISCUSSION.

Prof. S. A. Reeve.—I should like to refer to the statement made in the paper, quoting the remark of Mr. Campbell where he defines the gas-engine as "a cannon with its projectile fastened to the crank-shaft," * to say that that quotation voices a view of the mechanics of the gas-engine which I find to be quite common, and which I yet believe to be a totally mistaken view. The common idea is that when a gas-engine piston starts upon its stroke a blow is struck against the crank-pin. Now we know, from our simplest forms of construction, such as the hammer, that there must be velocity of motion destroyed in order to develop a blow; but in the gas-engine there is no such thing. At the moment of explosion no motion is developed, to be destroyed in striking the blow, and hence there can be no impact. What lost-motion there may have been in the engine-connections has already been taken up by the compression of the charge, and when the explosion-pressure is formed in the cylinder it comes on the piston-head decidedly more gently than it does in the case of the steam-engine; because in the steam-engine, when the valve is opened, the steam rushes in at a very high velocity, there *is* motion, which is destroyed against the piston, and there *is* a blow struck. But in the gas-engine there is no such thing. The pressure is developed, without motion, directly upon the piston-head, and the only reason why we have trouble with main-bearings, etc., is the fact of the enormous pressure. I think that if we attempted to design steam-engines working, within a single cylinder, from 400 pounds per square inch of initial pressure down to atmospheric pressure we should meet with very much more trouble than is found in the gas-engine.

* EDITOR.—The Author omitted this from his paper in revising it.

There is, thus, in gas-engines, a very great range of fluid-pressure during each stroke, and a corresponding need for allowance for it in designing them. This allowance, in Mr. Morgan's plan, has been accomplished by providing a very much increased weight of reciprocating parts, to take up, by its inertia, this excessive initial pressure. In addition to those parts present in the ordinary gas-engine: the piston and the primary, single-acting connecting-rod, there enters into this design the beam and the secondary, double-acting connecting-rod. In the first place, there are two pistons and primary rods, whereas the most

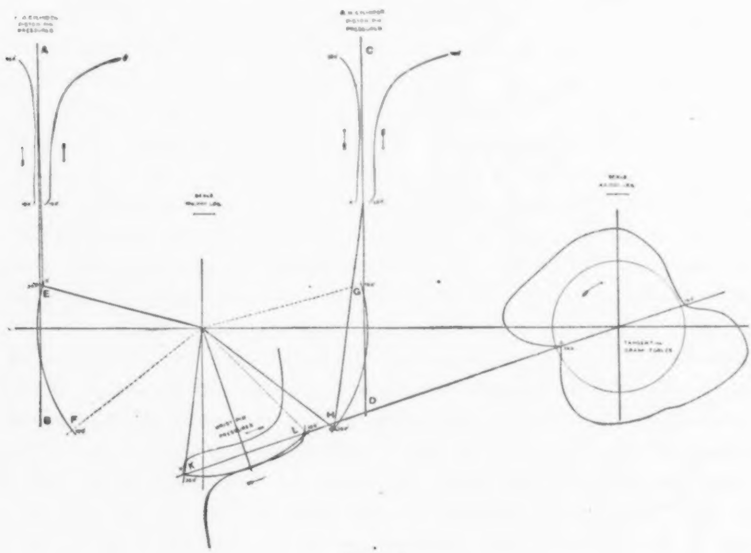


FIG. 82.

common design of gas-engine has only one of each. Secondly, the beam itself is quite a heavy piece of metal; but its moment of inertia is not so strikingly great as might appear at first sight to be the case; its radius of gyration is so small. Thirdly, the main connecting-rod does present a good deal of inertia, but no more so than any other connecting-rod for carrying the same amount of power. You will note the parts on this diagram Fig. 77.

The forces at work upon the different pins in the engine have been carefully worked out, and I can show you, from the results of this work, the uses of these inertia-forces for modifying the effective forces at work upon the crank-pin. They are displayed

in Fig. 82. At AB you see represented the center-lines of motion of the two pistons. AB is the axis of one cylinder; CD is the axis of the other. EF is the arc of motion for the first wrist-pin, and GH is the other. From an initial projected indicator card—which, I might say, was made the worst possible for the ends desired, with a perfectly vertical explosion-line rising to a peak some 50 pounds per square inch higher than would usually or properly occur—was calculated the effect of the combined forces upon the different pins. You will notice that there is no reversal of the forces except at dead centers, and that the forces are very much more constant on the crank-pin than they are in the original fluid-action upon the pistons.

The forces acting upon the main wrist-pin are shown by the curves KL. There is a very heavy initial force at the beginning of each stroke, but the peak shown there is more or less hypothetical in character and is much exaggerated over actual probability. During the rest of the stroke the pressure upon the pin, going in either direction, is fairly constant.

The diagram at the right shows the final net resultant forces at work upon the crank-pin, at each ten-degree division of its revolution, including fluid-pressure, inertia of all parts and the weight of the main connecting-rod. Those forces coming to the pin along the connecting-rod are shown by arrows in the direction of its axis. Compounded with them are the axial and vectoral inertia and the weight of the connecting-rod, to give the final resultant. In the diagram in the right hand upper corner are displayed the tangential effects of these same net resultants. In order to avoid confusion of lines their magnitude is displayed by radial arrows extending from the circumference outwardly; but their true direction is tangential, at the point whence the arrow rises.

I think that it has been urged that engines of an explosive type having heavy reciprocating parts cannot be run at high piston-speeds. This engine has a piston-speed of 810 feet per minute; but to show what it would do at even higher piston-speeds I have had projected here by Mr. Gifford a diagram showing the effective forces upon the crank-pin under a piston-speed of 1,000 feet per minute (exhibiting Fig. 83). This peak at the beginning of the stroke is, as before, due to more or less hypothetical conditions of operation. At this point in the stroke the pressure upon the crank-pin becomes almost zero.

This drop in pressure at this point might have been avoided by a modification in the design had it been one of the objects in view to attain a high piston-speed. Those ordinates are marked off at 10,000-pound intervals. This diagram has been gotten up to show that an engine of this type, with a very heavy triangular

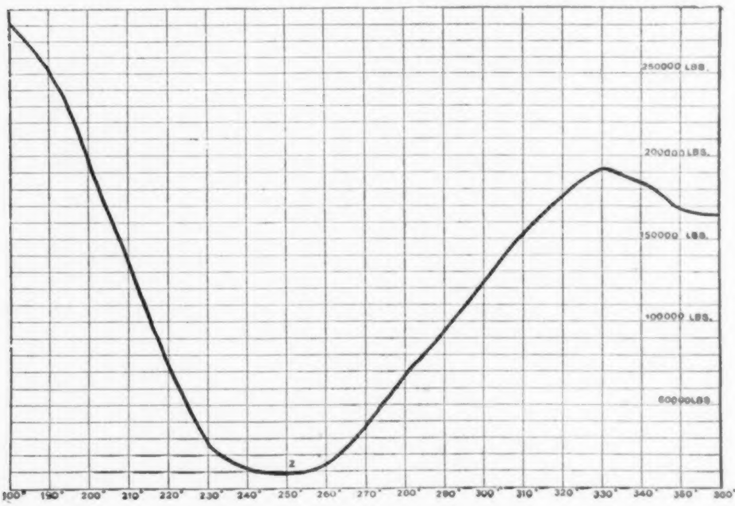


FIG. 83.

beam and three connecting-rods, can be operated at a piston-speed at high as 1,000 feet per minute without any reversal of the forces acting upon the crank-pin, except at or near dead-center.

Mr. B. H. Thwaite.—I had no idea that I should be called into a discussion on the subject of high power gas engines at this meeting. It has been impossible for me to read the paper that has been presented to you with the care it deserves; and, as I have promised your President that I would prepare a paper for your Society at your next meeting, I think you will perhaps appreciate what I have to say upon the subject of gas engines better if you wait until that time. Professor Reeve has already answered one of the main mistakes in this paper, namely, the impression that the gas engine is an explosive engine. It is nothing of the sort. In the early days of the free piston that statement might possibly have been correct. Now we obtain with gas of 100 to 120 B. T. U. a card which is quite different, and one that produces a more satisfactory crank-turning effect. Instead of

driving, direct on to the dead centre, I catch the maximum impulse about 20 degrees above the neutral line, giving remarkably good thermodynamic and uniform rotative results. The first card of the free piston engine using gas of 700 to 800 B. T. U., and no compression or dilution, does give the impression of an explosive engine, but now the modern gas engine is an essentially slow combustion one.

I notice that Mr. Campbell makes a suggestion, in the footnote of the first page of Mr. Morgan's paper, that "it is possible that exhaust gases from the gas engine can be profitably employed to heat the blast in the stoves of the blast furnace." I think the thermal value of the exhaust gases is so low as not to be worth anything. In our engine it is not much above 450 degrees Fahrenheit, and if you divide that temperature by half you will see that for all practical purposes it will be of no use for hot blast stoves of blast furnaces.

Mr. Morgan drew attention to the fact that Mr. Campbell says that the blast furnace gas is a primary producer of iron and also a gigantic gas producer. That is true. But the ordinary producer gives a gas of a thermal value of 150 to 200 degrees, whereas blast furnace gas has a thermal value of no more than 120 British thermal units, and sometimes as low as 80 B. T. U., a vast difference; indeed you must highly compress it to find its thermodynamic value. All you have to do is to compress it enough and then you will fire it with absolute or mechanical certainty.

I notice also that Mr. Morgan speaks about the difficulty of using blast furnace because of the dust it contains, which he truly says has been the bête noir of the gas engineer. Some nine years ago I killed this black beast and I have never had any trouble with it since. In every instance of the proper application of my system it has been a decided and continuous success.

As to the centrifugal system referred to by Mr. Morgan, that is and has always been more or less associated with my plant. Other people are simply trying to do with the fan, and some modification of my system, but if you do it that way you will not only decrease the thermal results appreciably, but you will have it to seriously increase the depreciation per cent. The Cockerill engine, which is being constructed now by the West Yorkshire Iron Company, England, is being equipped with

my system for treating and purifying the blast furnace gases. I would ask Professor Reeve if we are to understand that this design has ever been carried out practically, or is it merely an academic design?

Mr. Reeve.—It is already in construction.

Mr. Thwaite.—I am not going to criticize this design severely, because I always look with sympathy upon a new mechanical development—and I hope this one will have a thorough trial; I notice that Professor Reeve has already answered one point which occurred to me as a difficulty, and that is the question of the piston speed. When I saw this design I certainly thought he would have some difficulty in getting the ideal piston speed to secure the best thermal results. There is a certain piston speed, *i.e.*, 800 to 900 feet per minute, which enables you to get the best thermal results out of the gas engine; and if Professor Reeve tells me that you can get the speed, why, then you will get a good thermal effect.

Then I notice that he has in the design a vertical blowing cylinder. I have had a vertical blowing cylinder for my quarter crank blowing engine all the time, and this gas engine of mine works admirably and has given the greatest possible satisfaction. I work it at a speed of 120 revolutions, and I get a remarkably steady air pressure all the time. There is no more variation than one-eighth of an inch of the mercury gauge, and the engine works constantly night and day and for thirty-six days without a stop.

The final part of Mr. Morgan's paper, which commenced with a great deal of interesting philosophy—and that reminds me that you have in this meeting gentlemen who are really philosophers and who are really very witty, for yesterday you had scientific reasoning gilded with humor, especially in the paper presented by Mr. Barth. Finally, I find in Mr. Morgan's paper this statement: "If the wastes of the old generation are the profits of the new then large dividends are waiting us in the squandering of the past." I quite agree with that statement. I believe, gentlemen, that it is your duty as engineers to give some attention to this question of the internal combustion engine. I know you will be able to improve upon it, and you have here a subject that is deserving of your most serious attention; it is a tree of knowledge that deserves your care, and I believe you will eventually reap satisfactory fruit

from it. Mr. Morgan also speaks of the fearful power wastes of blast and coke furnaces. You must bear in mind that at present you cannot continue operations when the demand falls for pig iron, but with the new source of profit from the sale or use of power the blast furnaces may be kept in operation, which under the present conditions would be blown out. Certainly blast furnace gas is absolutely ideal for generating power by direct combustion, and is the best potential power next to waterfalls in existence.

Gentlemen, I thank you for your kind attention.

Mr. H. H. Suplee.—I desire to call attention to what appears to be an error in the paper regarding the action of the old explosive gas engine. Mr. Campbell described the gas engine as a gun whose projectile was connected to a crank shaft, but that statement is not quite correct if he was referring, as I suppose, to the old Otto & Langen engine. In that engine, the immediate predecessor of the modern four-cycle engine, the piston was not positively connected to the crank-shaft, the design being what is termed a "free piston" engine, the piston rod being a rack gearing into a pinion on the crank shaft and connected to it by a ratchet and pawl. The piston was undoubtedly a projectile, but when it was shot upwards it was not connected to the shaft, the power stroke being made by the downward stroke of the piston, this stroke being due to the atmospheric pressure on top of the piston, a vacuum being formed underneath after the explosion.

Mr. Thwaite.—I was only referring to the explosion on the piston. That is what I had reference to.

Mr. W. H. Morse.—I am interested in this discussion because it seems to advocate the vertical as against the horizontal cylinder. As a matter of history, I think the Westinghouse Company some two or three years ago developed a vertical engine of comparatively high power. If any Westinghouse men are present I should like to inquire why it is that of late in a number of papers we see illustrations of a 1,500 horse-power double acting tandem engine with horizontal rather than vertical cylinders.

Mr. E. A. Uehling.—There is no question before The Mechanical Engineers and the iron manufacturers of this country of greater importance than that of the utilization of blast furnaces gas in internal combustion engines. Some time ago I took the

pains to determine, as nearly as might be, the power that is now wasted in the manufacture of iron. The method of procedure and results obtained were published in the Stevens Institute Indicator in a paper called "The Blast Furnace as a Power Plant." Taking as near as possible the average practice of this country and the output of pig iron of 1902 as the basis of calculation, I found that there are in excess of one and three-quarters millions of horse power going to waste in the manufacture of pig iron in the United States. Since then I have had the opportunity of investigating the gases of several plants, and I find that my figures are too low. In my calculations I found that there were, over and above the power requirements of the blast furnaces themselves, and allowing sufficient gas for heating of the blast, 840 horse power per ton of iron produced per hour, available for outside purposes. In one plant, for instance, that I investigated where they operate four blast furnaces and a large steel works, if all the gas available, after deducting what is necessary to heat the blast, were utilized direct in efficient gas engines, there would remain over and above the 20,000 horse power about 18,000 horse power for sale. The available surplus power will differ at different plants because it depends on the class of iron made, the kind of fuel used and the character of ore smelted; but it is in no case less than 800 horse power per ton of iron manufactured per hour.

There seems to have been, especially in this country, great hesitancy in receiving this idea of utilizing blast furnace gas direct in gas engines, and on the first casual glance it would seem as though there could not be much in it, because it is a very lean gas, having only less than $\frac{1}{8}$ the calorific power of water gear; but, as Mr. Thwaite has just stated, it is none the less an ideal gas for internal combustion engines, and when you come to figure it through to the explosive mixture it is not a lean gas, *i.e.*, the heat effect does not differ so much in the explosive mixture in the engine as might be expected from the difference in calorific power. For instance, take an illuminating gas of a heat value of 764 British thermal units per cubic foot, when mixed with the theoretical air required for complete combustion it has only 101.8 heat units per cubic foot. An average blast furnace gas with only 87.5 British thermal units per cubic foot makes a theoretical mixture of 68.9 heat units per cubic foot. Thus we see that while the illuminating gas has nearly 9 times

the calorific power of the blast furnace gas, when mixed with sufficient air for complete combustion it has less than 1.5 the heat value of a similar mixture of the latter. Adding the necessary excess of air brings them still closer together. Illuminating gas requires nearly 10 times the air necessary for blast furnace gas; besides, the elements in illuminating gas do not lend themselves to high compression, because you are liable to get premature explosions; whereas the elements in blast furnace gas not only lend themselves, but it is a necessity that they should be highly compressed in order to make the explosion a certainty; this is largely due to the high percentage of CO_2, which is a great retarder of combustion.

Mr. F. R. Jones.—I wish a little more information; I wish we might know where this horse power is from—whether it is the heating power of the gas or the great power of the engine?

Mr. Uehling.—This rating is done by taking the heat units of the gas, determined by calorimeter and checked by analysis, and assuming an efficiency of twenty-five per cent. in the engine. Much better has been done than that. I believe it has been done up to thirty-two, and I think some tests have been made where thirty-five has been realized, but I have taken twenty-five per cent. as a basis, which I think is in every way conservative.

No. 1024.*

TESTS OF A COMPOUND ENGINE USING SUPERHEATED STEAM.

BY PROF. D. S. JACOBUS, HOBOKEN, N. J.
(Member of the Society.)

1. The engine on which the tests were made was located at the Millbourne Mills in Philadelphia. It is of the Rice and Sargent horizontal, cross-compound condensing type especially designed for using highly superheated steam. A Schmidt superheater is used for superheating the steam.

2. The tests were made to determine whether certain guarantees were fulfilled which were made by the builders—the Providence Engineering Works. The work was done conjointly by Mr. A. C. Wood and the writer, Mr. Wood representing the purchasers of the engine, and the writer the manufacturers. Mr. A. S. Vogt, Mechanical Engineer of the Pennsylvania R. R. Co., at Altoona, detailed two experienced men from his testing department to assist Mr. Wood, and with other observers and computers, furnished by Mr. Wood, and the staff of the writer, it was possible to check all important data by securing duplicate records. The writer wishes to state that he is greatly indebted to Mr. Wood for his valuable services and for his hearty co-operation in the work.

3. The high pressure cylinder of the engine was furnished with double beat poppet valves and the low pressure cylinder with Corliss valves. The inlet and exhaust valves of each cylinder were operated by separate eccentrics. The inlet valves were closed by vacuum dash pots and the exhaust by a positive motion. A portion of the highly superheated steam furnished to the engine passed through a coil in the receiver between the high and

---

* Presented at the New York meeting (December, 1903) of the American Society of Mechanical Engineers, and forming part of Volume XXV. of the *Transactions*.

the low pressure cylinders. There was a by-pass valve operated by the governor which regulated the amount of steam passing through the coil, a greater amount being made to pass through when the engine was heavily loaded than when it was lightly loaded. There were no steam jackets. The exhaust steam passed from the engine to a Worthington jet condenser connected with a Worthington vacuum pump.

4. The steam used by the engine was furnished by an Edge Moor water tube boiler. From the boiler it passed to the top of the Schmidt superheater which was set up at the side of the boiler. The saturated steam entering the superheater came in contact with the coldest gases as they left the superheater. From the upper coils the steam passed downward through a pipe outside of the superheater to the bottom of the lowermost set of coils of the superheater and thus, when only partly heated, was made to pass through the coils which were nearest the fire. This prevented the overheating of the lowermost coils of the superheater, which would have taken place had the steam been made to pass downward all the way through the superheater so as to bring the hottest steam near the fire. From the lowermost set of coils the steam passed upward and back and forth through the coils until it was finally led off from a header directly below the top set of coils and conveyed to the engine. Steam to drive the vacuum pump and for other purposes in the mill was furnished by a separate boiler.

5. The principal dimensions of the engine, measured when hot, were:

Bore of Cylinders	16.07 and 28.03 inches.
Length of Stroke	42 inches.
Diameter of Piston rods	3.00 and 3.50 inches.

6. The average clearance volumes computed from measurements made on the engine and from the working drawings were 4.1 per cent. for the high pressure cylinder, and 5.8 per cent. for the low pressure cylinder.

7. After the tests with superheated steam were completed a test was made with saturated steam. The writer was not present at this test but was represented by his associate, Prof. F. L. Pryor.

8. The principal results of the tests are given in detail in Table I. From this it may be seen that the water consumption per hour per indicated horse-power with superheated steam was 9.76

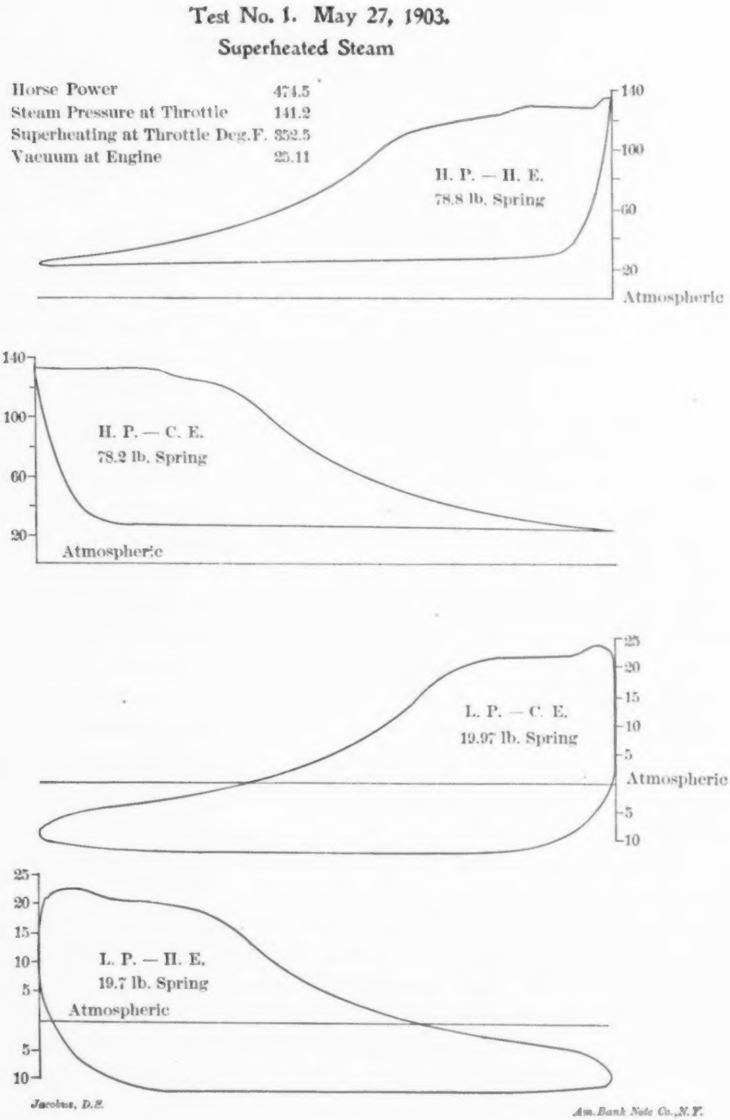

Test No. 1. May 27, 1903.
Superheated Steam
Horse Power 474.5
Steam Pressure at Throttle 141.2
Superheating at Throttle Deg. F. 352.5
Vacuum at Engine 25.11
H. P. — H. E.
78.8 lb. Spring
140
100
60
20
Atmospheric
H. P. — C. E.
78.2 lb. Spring
140
100
60
20
Atmospheric
L. P. — C. E.
19.97 lb. Spring
25
20
15
10
5
Atmospheric
5
10
L. P. — H. E.
19.7 lb. Spring
25
20
15
10
5
Atmospheric
5
10
Jacobus, D.S.
Am. Bank Note Co., N.Y.

FIG. 84.

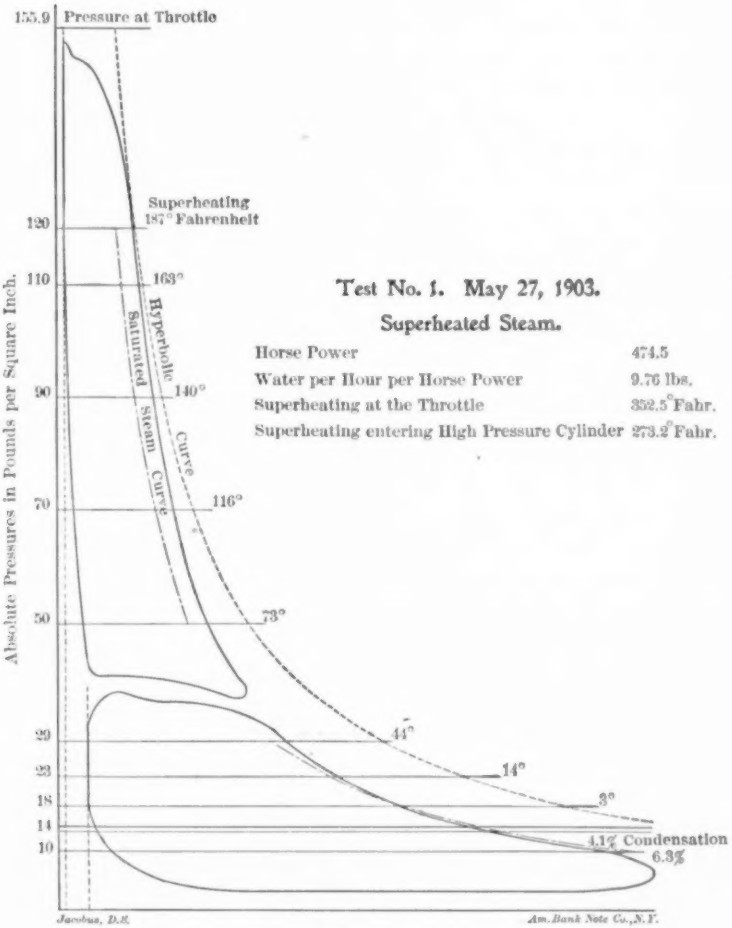


FIG. 85.

pounds at 474.5 horse-power, 9.56 pounds at 420.4 horse-power, and 9.70 pounds at 276.8 horse-power. The pressure of the steam at the engine throttle was slightly over 140 pounds per square inch, and the superheating at the engine throttle from about 350 degrees to 400 degrees Fahr. The vacuum measured near the engine ranged from about 25 to 26 inches of mercury. The heat consumption per minute per indicated horse-power, according to the standard recommended by the Civil Engineers of London, where the engine is charged with the heat in the

steam at the throttle valve and is credited with returning the feed water to the boiler at the maximum possible temperature that could be obtained by a feed water heater in the exhaust pipe, was 205.0 British Thermal Units at 474.5 horse-power, 203.7 British Thermal Units at 420.4 horse-power, and 208.8 British Thermal Units at 276.8 horse-power.

9. With saturated steam the water consumption was 13.84 pounds per hour per horse-power at 406.7 horse-power, and the heat consumption 248.2 British Thermal Units per minute per indicated horse-power. In obtaining this figure the steam condensed through radiation of the steam pipe leading from the boilers to the engine has been deducted in order to make the figures for the heat and coal consumption strictly comparable with those given for the superheated steam, which are based on the temperature of the superheated steam at the engine throttle valve. In this test the vacuum was poorer than in any of the others, being 24.5 inches. It must also be borne in mind that the engine was built for superheated steam, and the cylinder ratio was not what the makers would use for saturated steam. In computing the heat consumption for this test an allowance is made for the drip from the high pressure steam main at the coil in the receiver. This drip amounted to 280 pounds per hour over what could be accounted for by the radiation of the steam main from the boiler to the engine. This amount of the high pressure steam was, therefore, condensed in the receiver coil, and imparted heat to the receiver.

10. Tests were made to determine the economy of the boiler and of the superheater. The coal used was run of mine, Georges Creek, Cumberland. The equivalent evaporation from and at 212 degrees Fahr. per pound of dry coal for the boiler alone was 10.30 pounds, for the superheater it was 8.97 pounds, and for the boiler and superheater combined it was 10.07 pounds. The heat of combustion of the coal determined by means of a Mahler bomb calorimeter was 14,060 British Thermal Units per pound. The efficiency of the boiler based on the heat of the coal was 70.7 per cent., of the superheater 61.6 per cent., and of the combined boiler and superheater 69.2 per cent. In computing the efficiency of the superheater the specific heat of the superheated steam was taken at the value found by Regnault for atmospheric pressure or .48. At higher pressures this value is probably too low. On the other hand the amount of superheat-

ing given by the mercury thermometers used in the tests is greater than that which would have been given by an air thermometer, and this introduces an error in the opposite direction from that which probably exists through taking the specific heat of the steam at .48 and the one error therefore, tends to counterbalance the other.

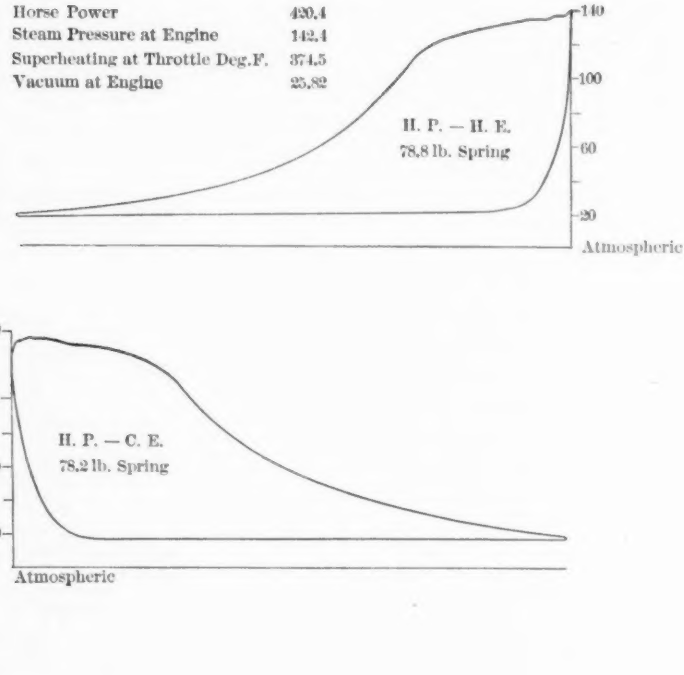
11. The coal consumption may be obtained directly from the heat consumption of the engine already given by multiplying by 60 and dividing by 9,725 for the superheated steam tests, and by 9,947 for the test with saturated steam, 9,725 being the heat imparted to the boiler and the superheater in British Thermal Units per pound of coal burned in the tests with superheated steam and 9,947 that imparted to the boiler in the test with saturated steam. When computed in this way any loss of heat by radiation in the pipe leading from the boiler to the engine is eliminated, which is a necessary condition in the present case as the pipe is exceptionally long. On this basis the coal consumption per hour per indicated horse-power for superheated steam was 1.265 pounds at 474.5 horse-power, 1.257 pounds at 420.4 horse-power, and 1.288 pounds at 276.8 horse-power. For the test with saturated steam the coal consumption was 1.497 pounds per hour at 406.7 horse-power. In the figures just given for the coal consumption the coal used to drive the independent vacuum pump is not included. The figures, therefore, represent what would have been obtained per indicated horse-power if the vacuum pump had been driven from the main engine.

12. The coal consumption forms a more accurate basis for comparing the efficiencies of engines using saturated and superheated steam than the heat consumption, because when the coal consumption is obtained it allows for any difference in the economy of the combined boiler and superheater from that of an ordinary boiler. Furthermore, the results obtained for the coal consumption are practically independent of the value employed for the specific heat of the steam which is not the case with the heat consumption. For example, if the specific heat was taken at .6 instead of .48 in computing the results which have been given it would cause a difference in the coal consumption of less than one-half of one per cent., whereas it would cause a difference in the heat consumption of about 3½ per cent.

13. In the test with saturated steam the coal consumption was about 19 per cent. greater than that obtained at the same power

Test No. 2. June 19, 1903.
Superheated Steam.

Horse Power	420.4
Steam Pressure at Engine	142.4
Superheating at Throttle Deg.F.	374.5
Vacuum at Engine	25.82



Jacobus, D.E. *Am.Bank Note Co.,N.Y.*

FIG. 86.

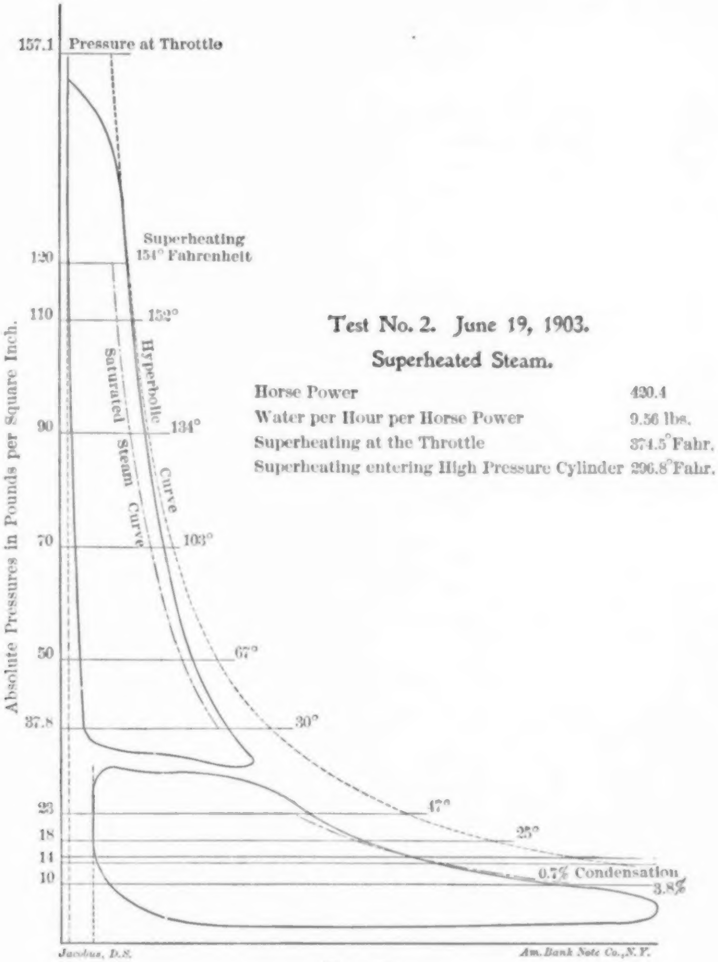


FIG. 87.

with superheated steam. This does not form a proper basis of comparison, however, for the two reasons already stated, which are that the engine was not built for saturated steam and that the vacuum in the test was not as high as it should have been. An engine of the same general type built for saturated steam, but of about twice the power, was tested by the writer at the Brooklyn plant of the American Sugar Refining Company and the results were embodied in a paper presented by him at the Saratoga meeting. The most economical heat consumption obtained

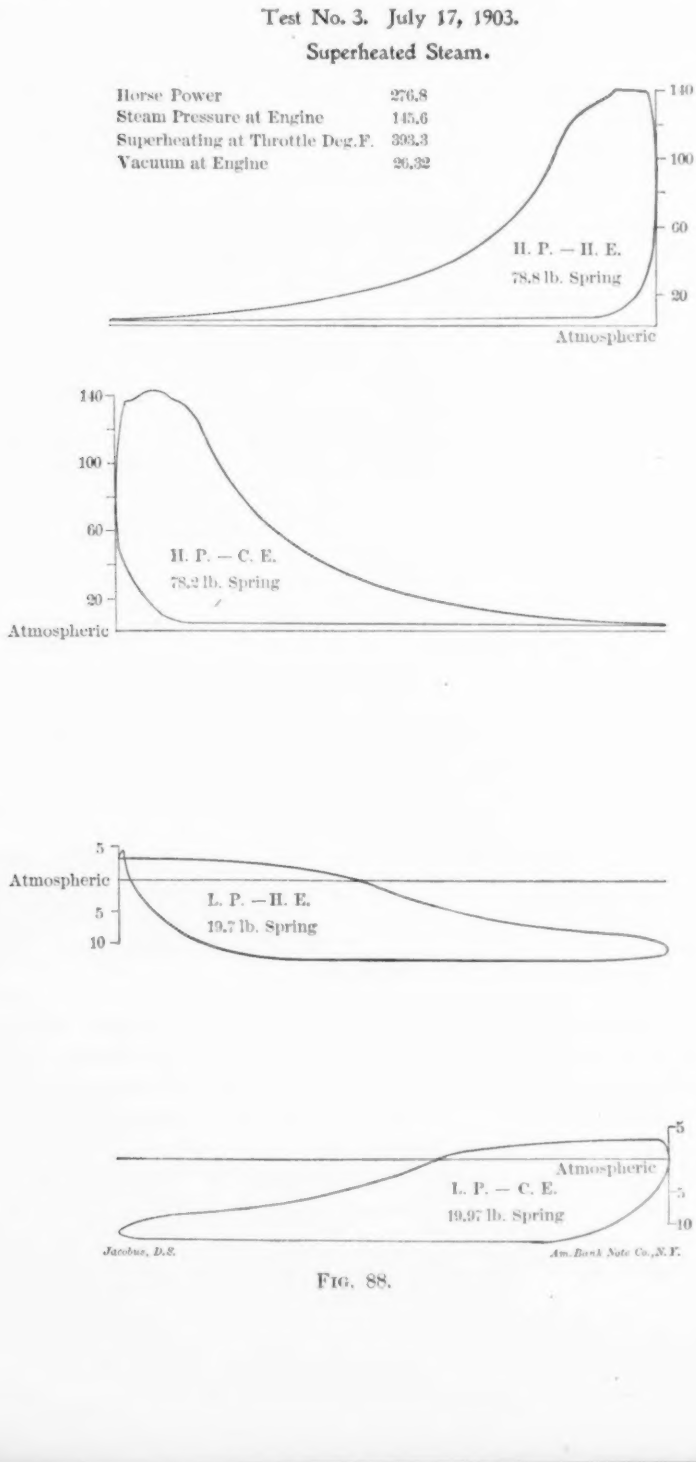

Test No. 3. July 17, 1903.
Superheated Steam.
Horse Power 276.8
Steam Pressure at Engine 145.6
Superheating at Throttle Deg.F. 393.3
Vacuum at Engine 26.32
H. P. — H. E.
78.8 lb. Spring
Atmospheric
H. P. — C. E.
78.2 lb. Spring
Atmospheric
L. P. — H. E.
19.7 lb. Spring
Atmospheric
L. P. — C. E.
19.97 lb. Spring
Atmospheric
Jacobus, D.S.
Am. Bank Note Co., N.Y.

Fig. 88.

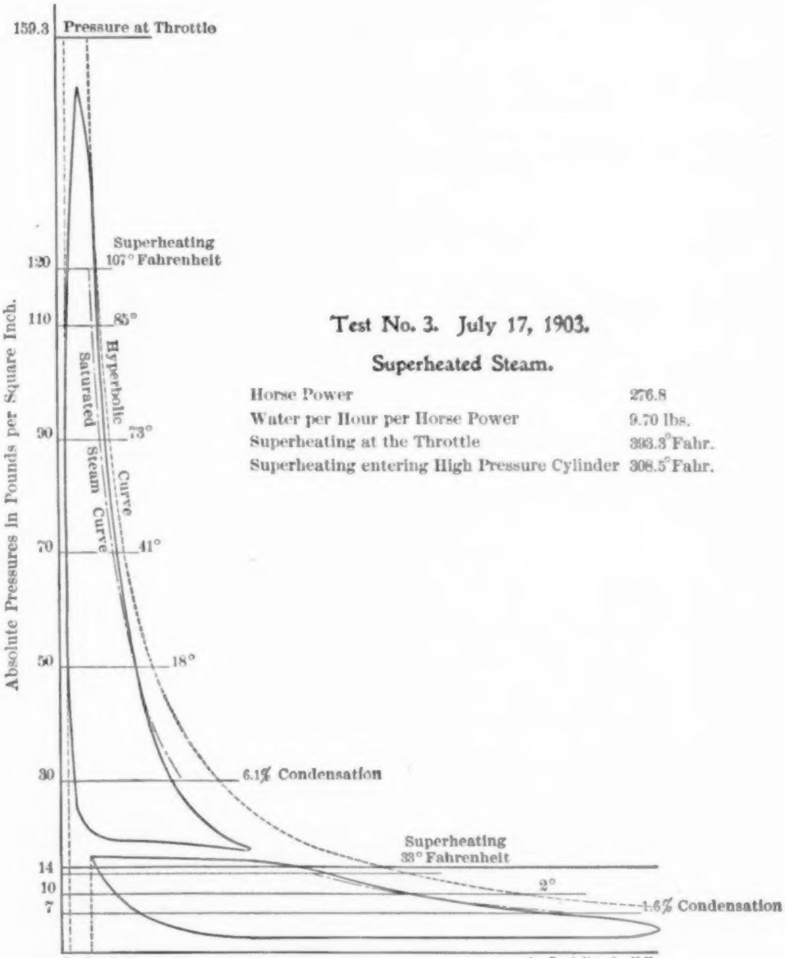


FIG. 89.

was 222.7 British Thermal Units per minute per horse-power. The vacuum in this case was exceptionally high, being over 28 inches. If the vacuum had been about 26 inches, as in the present tests, it is probable that the heat consumption would have been about 230 British Thermal Units per minute per indicated horse-power, and the coal consumption on the basis already given 1.42 pounds per hour per horse-power. As the minimum coal consumption with the engine using superheated steam was 1.257 pounds per hour per horse-power there is a gain in favor of using

Test No. 4. July 24, 1903.

Saturated Steam.

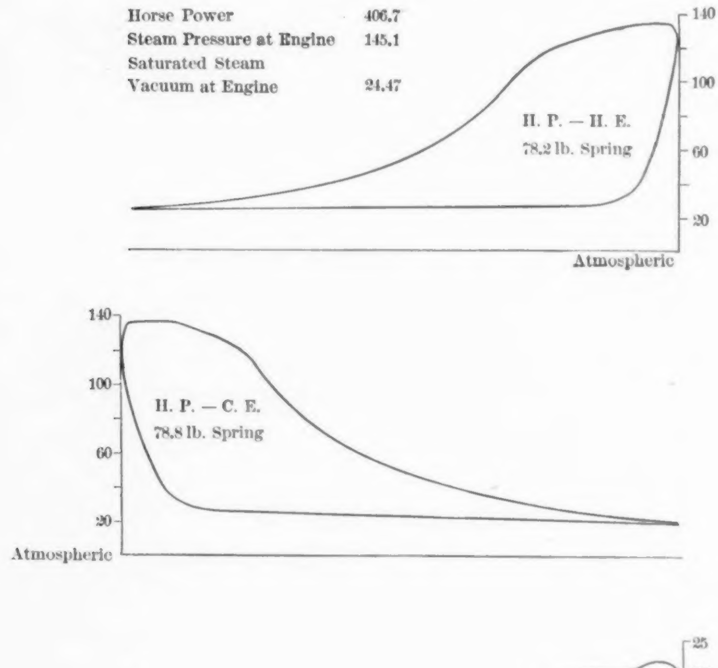


FIG. 90.

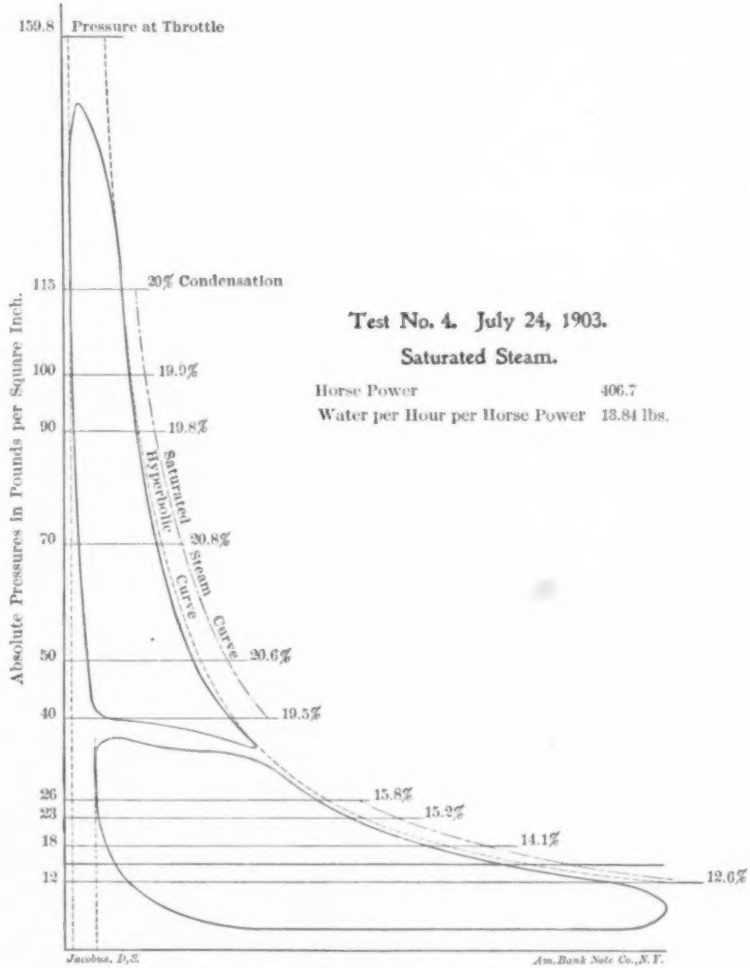


FIG. 91.

superheated steam of .163 pounds per hour per horse-power, or the larger steam engine using saturated steam is about 13 per cent. less economical than the smaller one using superheated steam.

Methods of Conducting the Tests.

14. The water was weighed in an iron tank of about 1,000 pounds capacity placed on a platform scale, and it was then emptied into a lower tank connected with the suction pipe of a

geared boiler feed pump driven by the main engine. All steam and water connections, where there was a possibility of leakage occurring, were either broken or blanked off.

15. Indicators of the Star Brass Manufacturing Co., having outside springs, were used on the high pressure cylinder. Tabor indicators of the ordinary pattern were used on the low pressure cylinder.

16. Mercury columns were used to measure the vacuum both in the exhaust pipe near the engine and near the condenser. A spring vacuum gauge was also used near the condenser which was calibrated in order to determine its correction.

17. Mercury thermometers placed in thermometer wells were used for determining the temperatures. Where the temperatures were not too high mercury was used in the wells, but if the mercury evaporated, melted solder was employed. In every case the error due to exposing a portion of the stem of the thermometer was determined by noting the graduation to which it was immersed and estimating the temperature of exposed part of the mercury in the stem by tying a second thermometer against the stem.

18. Readings were taken at intervals of 15 minutes throughout the tests. The water record was balanced up at the end of each hour. The feed pump was kept running uniformly and was not stopped in balancing up the water. The engine was run for some time before starting each of the tests in order that the temperatures should become practically constant. The steam leakages were carefully estimated, and in one case a leakage test was made on the boiler and the results confirmed the amount which was estimated. The steam leaks amounted to 40 pounds per hour as a maximum.

19. The engine test of June 19th was run at the same time as the tests of the boiler and superheater. Arrangements were made for a continuous engine test of twelve hours or more, but six hours after starting the engine test the water rate suddenly increased. It was found that a gasket at the receiver head which had been leaking slightly during the first part of the test, and the leakage of which had been estimated at three pounds per hour, was leaking much more than at first. It was impossible to estimate the amount of the increased leakage, and the engine test was therefore discontinued at the end of six hours. As the leak did not affect the tests of the boiler or superheater these

tests were made of 14 hours duration. The readings of the amount of feed water indicated uniform conditions during the engine test, the average rate per hour for the first two hours being 4,026 pounds; for the second two hours, 4,012 pounds, and for the last two hours of the test, 4,016 pounds. This shows that the duration of the test was ample for securing accurate results.

20. The mean effective pressures and indicated horse-powers are given in Table II. Table III gives the average data and results of the boiler test, Table IV those for the superheater test and Table V the data used in computing the superheating or the percentage of condensation existing in the cylinder at various points in the stroke. The average indicator cards and the combined diagrams for each of the tests are shown in Figs. 1 to 8 inclusive. The amount of superheating or of priming existing at various points in the expansion lines is marked on the combined diagrams. Zeuner's formula for superheated steam is used in computing the amount of superheating.

TABLE I.

Average Data and Results of Engine Tests.

1. Date of Test: 1903	May 27	June 19	July 17	July 24
2. Condition of Steam: superheated or saturated	Sup.	Sup.	Sup.	Sat.
3. Length of tests in hours and decimals of an hour	5.15	6.033	6.00	6.50
4. Water fed to boiler in pounds per hour corrected for leakage	4633	4018	2684	5630
Average Pressures.				
5. Steam at boiler in pounds per square inch above atmosphere	147.0	147.9	148.5	149.3
6. Steam near throttle in pounds per square inch above atmosphere	141.2	142.4	145.6	145.1
7. Steam in receiver in pounds per square inch above atmosphere	23	17	3	22
8. Vacuum near engine in inches of mercury	25.11	25.82	26.32	24.47
9. Vacuum at condenser in inches of mercury	25.80	26.79	26.81	25.24
10. Barometric pressure in inches of mercury	30.16	29.80	30.01	30.00
Temperatures of Steam in Degrees Fahr. by Mercury Thermometers.				
11. Leaving the superheater	766.4	808.0	849.0	....
12. At the engine throttle	713.7	736.3	756.8	....
13. Entering the high pressure cylinder	634.4	658.6	672.0	....
14. Leaving the high pressure cylinder	346.5	331.5	287.7	262.0
15. Entering the low pressure cylinder	408.0	395.9	353.6	269.1
16. Leaving the low pressure cylinder	135.1	128.2	124.1	....
Amount Steam was Superheated in Degrees Fahr.				
17. Leaving the superheater	402.4	443.4	384.1	....
18. At the engine throttle	352.5	374.5	393.3	....
19. Entering the high pressure cylinder	273.2	296.8	308.5	....
20. Leaving the high pressure cylinder	84.7	77.0	52.0	....
21. Entering the low pressure cylinder	146.2	141.4	117.9	7.1

Horse-power and Economy.				
22. Revolutions per minute	103.28	102.34	102.49	102.20
23. Indicated horse-power developed by the engine	474.5	420.4	276.8	406.7
24. Water consumption in pounds per hour per indicated horse-power	9.76	9.56	9.70	13.84
25. Maximum temperature to which the feed water could be heated by the exhaust of the engine in degrees Fahr.*	133.5	125.1	121.7	148.4
26. Heat consumption in British Thermal Units per minute, corresponding to the heat in steam at the throttle valve, and the maximum temperature of feed water—Standard of the Civil Engineers of London	205.0	203.7	208.8	248.2
27. Heat imparted to the boiler and the superheater in British Thermal Units per pound of dry coal	9725	9725	9725	9947
28. Coal in pounds per hour per horse-power on basis of the heat consumption recommended as a standard by the Civil Engineers of London. In this the loss due to radiation of the steam pipe from the boiler to the engine is not charged against the engine, and it is credited with the maximum temperature at which it can return the feed water to the boiler	1.265	1.257	1.288	1.497
29. Actual water evaporated and superheated in pounds per pound of dry coal in test made June 19	....	7.09	....	....
30. Actual coal burned in pounds per hour per indicated horse-power	....	1.348	....	....

* In the fourth test, which was made with saturated steam, allowance is made for the heating effect of drip water drawn from the receiver, which amounted to 280 pounds per hour.

TABLE II.

Mean Effective Pressures and Indicated Horse-powers.

Constant for high pressure cylinder, 0.04227, and for low pressure cylinder, 0.1299.

No. of Test.	High Pressure Cylinder.					Low Pressure Cylinder.						Indicated Horse-power.		
	Head End.		Crank End.		Average for two Ends.	Head End.		Crank End.		Average for two Ends.	Revolutions of Engine per Minute.	High Pressure Cylinder.	Low Pressure Cylinder.	Total.
	Scale.	Average M. E. P.	Scale.	Average M. E. P.		Scale.	Average M. E. P.	Scale.	Average M. E. P.					
1	78.8	51.60	78.2	51.48	51.54	19.7	18.34	19.97	18.87	18.60	103.28	225.0	249.5	474.5
2	78.8	47.71	78.2	52.00	49.85	19.7	15.39	19.97	15.39	15.39	102.34	215.8	204.6	420.4
3	78.8	38.88	78.2	38.70	38.79	19.97	8.05	19.7	8.29	8.17	102.49	168.0	108.8	276.8
4	78.2	40.66	78.8	39.19	39.93	19.97	17.67	19.7	17.63	17.65	102.20	172.4	234.3	406.7

TABLE III.

Average Data and Results of Boiler Test.

Made June 19 *and* 20, 1903.

1. Size of grate; length, 7 feet; breadth, 9 feet, 6.5 inches; area, square feet		66.8
2. Water heating surface in square feet		3,332.
3. Superheating surface in square feet		350.
4. Ratio of total heating surface to grate surface		55.1:1
5. Horse-power of boiler as rated by builders		370.
Total Quantities.		
6. Length of test, hours		14
7. Weight of moist coal burned during test, in pounds.	Total fired	7,205.
	Correction due to thickness of fire at end of test	158.
	Corrected weight	7,047.
8. Weight of dry coal burned during test, in pounds		6,823.
9. Total weight of water evaporated during test, in pounds		58,585.
10. Equivalent weight of water that would have been evaporated from and at 212 degrees Fahr., in pounds		70,302.
11. Weight of ashes, in pounds		717.
Average Quantities.		
12. Steam pressure in pounds per square inch above atmosphere		147.4
13. Temperature of feed water in degrees Fahr		66.6
14. Temperature of flue gases in degrees Fahr.	Actual reading	569.
	Corrected reading	582.
15. Temperature of boiler room in degrees Fahr.	Wet bulb	66.1
	Dry bulb	71.6
16. Draught in inches of water	In back connection	0.1
	Over the fire	0.1
17. Percentage of moisture in coal		3.18
18. Percentage of ash, based on dry coal		10.51
19. Barometer, in inches of mercury		29.82
Economic Results.		
20. Factor of evaporation		1.200
21. Weight of water evaporated per pound of fuel, including moisture, in pounds	Actual	8.31
	From and at 212 degrees Fahr	9.99
22. Weight of water evaporated per pound of dry fuel, in pounds	Actual	8.59
	From and at 212 degrees Fahr.	10.30
23. Weight of water evaporated per pound of combustible, in pounds	Actual	9.59
	From and at 212 degrees Fahr.	11.51
24. Weight of dry coal burned per hour per square foot of grate surface, in pounds		7.30
25. Calorific value of the dry coal in British Thermal Units per pound		14,060.
26. Calorific value of the combustible in British Thermal Units per pound		15,510.
27. Efficiency of boiler (based on the combustible), in per cent		71.7
28. Efficiency of boiler including the grate (based on the dry coal), in per cent		70.7
29. Water evaporated, in pounds per square foot of total heating surface per hour, from and at 212 degrees Fahr		1.36
30. Average horse-power developed based on standard recommended by A. S. M. E., viz., 34.5 pounds of water evaporated from and at 212 degrees Fahr., per hour		145.7

TABLE IV.

Average Data and Results of Superheater Test.

1. Heating surface in square feet	642.
2. Grate surface in square feet	4
3. Duration of test in hours	14
4. Pressure of steam furnished to superheater in pounds per square inch above the atmosphere	147.4
Temperatures in Degrees Fahr. by Mercury Thermometer.	
5. Steam entering the superheater	365.6
6. Steam leaving the superheater	809.1
7. Amount steam was superheated	443.5
Total Quantities.	
8. Steam passing through the superheater in pounds	58,025.
9. Heat imparted to the steam in British Thermal Units	12,352,000.
10. Total moist coal fired, in pounds	1,473.
11. Total dry coal consumed in pounds	1,426.
12. Coal burned per square foot of grate surface per hour in pounds	25.5
13. Heat imparted to the steam in the superheater per pound of coal burned, in British Thermal Units	8,662.
14. Equivalent evaporation from and at 212 degrees Fahr. in pounds per pound of dry coal	8.97
15. Heat of combustion of the dry coal in British Thermal Units per pound	14,060.
16. Efficiency of the superheater based on the heat of combustion of the coal in per cent	61.6
Combined Economy of the Boiler and Superheater.	
17. Actual evaporation of the boiler per pound of dry coal in pounds	8.586
18. Coal burned in superheater in pounds per pound of steam passing through it	.02458
19. Coal burned in superheater in pounds per pound of coal burned at boiler, Item 17 × Item 18	.211
20. Actual evaporation of the combined boiler and superheater, Item 17 ÷ (1 + Item 19)	7.090
21. Factor of evaporation of the combined boiler and superheater	1.421
22. Equivalent evaporation in pounds per pound of dry coal, from and at 212 degrees Fahr., for the combined boiler and superheater	10.07
23. Efficiency of the combined boiler and superheater based on the heat of combustion of the coal, in per cent	69.2

TABLE V.

DATA USED IN COMPUTING THE AMOUNTS OF SUPERHEATING AT VARIOUS POINTS IN THE EXPANSION LINE OF INDICATOR DIAGRAMS. AVERAGE FOR SIX DIAGRAMS.

Weights of Steam per Stroke in Pounds.

Date of Test	May 27	June 19	July 17	July 24
Passing through cylinders of engine	0.3739	0.3272	0.2182	0.4362
Retained in clearance space of high pressure cylinder	0.0619	0.0506	0.0299	0.0693
Retained in clearance space of low pressure cylinder	0.0345	0.0319	0.0357	0.0489
Total contained in high pressure cylinder during expansion	0.4358	0.3778	0.2481	0.5055
Total contained in low pressure cylinder during expansion	0.4084	0.3591	0.2539	0.4851

TABLE V.—*Continued.*

Date of Test.	Condition of steam superheated or saturated.	Cylinder.	Average length of indicator cards in inches.	Absolute pressure in lbs. per sq. inch.	Length including clearance, to the expansion curve, in inches.	Volume occupied in cylinder by 1 lb. of steam, in cu. ft.	Temp. in deg. Fahr. for steam at the same pressure and volume by Zeuner's formula.	Amount of superheating in degree Fahr. or per cent. of condensation.
1	2	3	4	5	6	7	8	9
May 27	Superheated.	H.P.	4.97	120	2.09	4.67	528	187 deg.
"	"	"	"	110	2.21	4.94	497	163 "
"	"	"	"	90	2.60	5.81	460	140 "
"	"	"	"	70	3.20	7.16	419	116 "
"	"	"	"	50	4.14	9.28	354	73 "
"	"	L.P.	"	29	2.03	14.88	292	44 "
"	"	"	"	23	2.42	17.74	249	14 "
"	"	"	"	18	2.99	21.92	225	3 "
"	"	"	"	14	3.61	26.46	....	4.1 p.c.
"	"	"	"	10	4.83	35.41	....	6.3 "
June 19......	"	H.P.	4.18	120	1.47	4.50	495	154 deg.
"	"	"	"	110	1.59	4.88	487	152 "
"	"	"	"	90	1.88	5.76	454	134 "
"	"	"	"	70	2.30	7.06	408	105 "
"	"	"	"	50	3.00	9.20	348	67 "
"	"	"	"	37.8	3.71	11.37	294	30 "
"	"	L.P.	4.16	23	1.87	18.63	282	47 "
"	"	"	"	18	2.28	22.71	247	25 "
"	"	"	"	14	2.75	27.39	....	0.7 p.c.
"	"	"	"	10	3.65	36.36	....	3.8 "
July 17......	"	H.P.	4.18	120	0.91	4.25	448	107 deg.
"	"	"	"	110	0.96	4.48	420	85 "
"	"	"	"	90	1.14	5.33	393	73 "
"	"	"	"	70	1.38	6.45	344	41 "
"	"	"	"	50	1.83	8.56	298	18 "
"	"	"	"	30	2.71	12.66	....	6.1 p.c.
"	"	L.P.	"	14	2.08	29.16	242	33 deg.
"	"	"	"	10	2.72	38.14	195	2 "
"	"	"	"	7	3.72	52.16	....	1.6 p.c.
July 24......	Saturated.	H.P	4.17	115	1.33	3.05	....	20.0 "
"	"	"	"	100	1.52	3.49	....	19.9 "
"	"	"	"	90	1.67	3.84	....	19.8 "
"	"	"	"	70	2.10	4.82	....	20.8 "
"	"	"	"	50	2.88	6.62	....	20.6 "
"	"	"	"	40	3.60	8.27	....	19.5 "
"	"	L.P.	4.18	26	1.77	12.99	....	15.8 "
"	"	"	"	23	2.00	14.68	....	15.2 "
"	"	"	"	18	2.55	18.71	....	14.1 "
"	"	"	"	12	3.80	27.89	....	12.6 "

DISCUSSION.

Mr. George I. Rockwood.—I have not seen an advance copy of this paper by Professor Jacobus, so in what I have to say I cannot attempt to speak with any preparation at all. The figures speak for themselves, however, and suggest some questions. For instance, in two tests—numbered 2 and 4—one with and the other without superheat, the load being substantially alike in each case—being 420 horse-power with superheat, and 406 with saturated steam—the steam consumption per horse-power per hour is given as 9.5 pounds and 13.8 pounds respectively. The difference represents an apparent saving of over 30 per cent. due to superheating, which is so much greater than usual as to need explanation. Now, as the paper deals with the economy of steam obtained by the use of superheaters, it may be in order to discuss other sides of the general question of the actual economy of coal effected by their use.

A factor which greatly affects their value is the manner in which the regulation of the superheat is accomplished. To illustrate, I lately visited a turbine station in Rhode Island and saw there one of these independently-fired superheaters at work, and in this case the upper doors were just a little open and the ash-pit doors open too; this was to prevent too hot a fire, as with the fire doors closed the temperature of the steam would be too hot for safe use.

Another case of the same sort of temperature regulation is found in the provision of a hollow bridge wall and a register on the outside face of the brick setting, for the purpose of cooling the gases as they come from the fire. It is obvious that such means of temperature regulation involve a loss of heat out of proportion to any saving effected in the engine or turbine. Its use is permissible only in cases where the steam exhausted from an engine is to be used in manufacturing processes after being superheated.

The most economical way, and, indeed, the only economical way, to preserve the temperature of the steam at a constant point is to keep the weight of steam flowing through the superheater proportionally the same as the weight of coal burnt on the superheater grates. One way I have employed of getting around the difficulty is to provide a " bleeder " pipe at the engine throttle

and to open it when the load is light or the engine shut down. In this case there are 3,000 horse-power of boilers and only ten per cent. of the steam is made at a high pressure—200 pounds per square inch—for use in the engine. The rest is used at 100 pounds pressure in heating water in a dye-house. The "bleeder" simply diverts some of the steam intended for use in the engine into the dye-house pipes after it has passed through the superheater. In any bleachery or dye-house or steam-driven paper mill such a method of regulation would be effective. The thing to be avoided is too hot a fire under the superheater with partial load, as this results in burning out the superheater tubes and ruining the engine or turbine.

Mr. R. H. Rice.—It may be interesting to give some further particulars of the construction of this engine which is utilizing very high temperatures of steam without difficulty—in fact, with no more trouble than any of the ordinary saturated steam engines of the same builders.

The double beat poppet valves have seats surrounded by the inlet steam in such a way that the expansion of the seat is equal in extent and effect to that of the valve—thus overcoming completely the characteristic defect of ordinary designs of their type of valve, namely, excessive leakage at any temperature other than the particular one at which they were originally ground. The inlet valves are driven by the ordinary trip gear of the builders, with vacuum dashpots, with the addition of a simple linkage which controls the closure of the valve independent of the extent of closing motion imparted by the dashpot, and thus prevents slamming or partial closure of the valve. The exhaust valves are actuated by a system of links devoid of cams, always in connection with the eccentric except when hand-actuated at starting or stopping, and which keeps the valve stationary during the forward stroke, as is necessary when using the poppet type, and all joints are adjustable for wear.

The stuffing-boxes are on long necks to take them well away from the superheat, and the piston-rod stuffing-boxes have metallic packing provided with water jackets which, however, have never been used. The piston packing consists of two simple cast-iron spring rings with joint plates.

The horse power cylinder is so designed that the working portion of its band is a simple cylinder without ribs, all connections to the cylinder, such as valve cleets, lagging bosses, inlets

and exhaust gauges, etc., being at the ends. The clearance of the poppet valve cylinder has been kept fully as small as the equivalent Corliss-valve cylinder, in contrast with foreign practice, where it is usually much greater.

The only trouble noticed with lubrication was a smoking due to the carbonization of the animal or vegetable constituents of the original oil used. On notifying the oil makers of this trouble they at once produced an oil which eliminated all complaint.

The operation of the superheater has proved to be simple: in fact it is easier to run than a boiler, since the pyrometer dial is the only thing needing to be watched. Fire is never built in the superheater without a flow of steam through the coils, under which conditions there is no sign of deterioration. The temperature is readily regulated, even when the engine is shut down for changes in the mill, which happens once or twice in 24 hours in regular operation. If the shut-down is for more than a few minutes a small flow of steam is secured by cracking the throttle valve and allowing a little steam to blow through the engine, but for short stoppages this is not necessary. The pipes, cylinders and receiver are covered with 3 inches of a standard magnesia covering over pipes and flanges.

Lines 13, 14, 15, Table I, page 6, of the paper shows interestingly the control of the temperatures of steam entering the two cylinders by the governor-controlled valve of the reheater, test No. 3, with light load, showing an inlet temperature of 672 degrees to high pressure cylinder and 353 degrees to low pressure cylinder, while test No. 1, with heavy load, shows 634 degrees to high, and 408 degrees to low. This variation of temperature inversely with load in high cylinder is believed to be necessary with high superheat to protect it from burning the oil. In this engine the regulation is effected by giving to the low pressure cylinder the superheat which has been taken out of the steam going to the high cylinder.

Although the cylinder ratio is different from that usually adopted in this country for saturated steam, it nevertheless corresponds with European practice, and considering the low vacuum and the absence of jacket, the steam consumption found —13.84 pounds—is not an excessive figure.

In comparing the coal consumption of this engine and that reported by Professor Jacobus at the Saratoga meeting, it should be noticed that the best heat consumption of the latter engine

was obtained with so low a m. e. p. that the engine, if proportional for this m. e. p. at full load would be excessive in cost on account of its considerable cylinder dimensions; so much so as to make its cost comparable to that of a superheated steam engine and its superheater complete, whereas the superheated steám engine shows a saving of about 12 per cent. of coal, which is therefore obtained with only a small additional investment.

Tests of engine constructed on this system in England with a vacuum of about 28 inches of mercury show a steam consumption of 9 pounds per indicated horse power, and it is probable with a load of about 375 horse power (which could not be obtained in this case owing to the arrangement of the mill), and an equally good vacuum, that the same figure could have been obtained in this case.

In regard to the superheater at Newport, I am somewhat familiar with that installation. The superheater was made of sufficient capacity to superheat the steam for 2,000 kilowatt capacity of turbines, and it is now superheating steam for less than 200 kilowatts capacity. It is, therefore, operating under entirely abnormal conditions.

Mr. E. T. Child.—I would like to ask Professor Jacobus what was the ratio of cylinders in the Brooklyn engine? And is this ratio of 3 to 1 in the Millbourne Mills engine the blast for superheated steam? It is often desirable to use superheated steam in engines already built and with cylinder-ratios adapted for saturated steam. Will these larger cylinder-ratios make the engine illy adapted for superheated steam?

Professor Jacobus.—What is your last question? I did not quite catch it.

Mr. Child.—Is this ratio of 3 to 1 distinctly based on the use of superheated steam, and, if you have an engine built for saturated steam, say, 4 or 5 or 6 to 1, is that illy adapted for use with superheated steam?

Professor Jacobus.—The ratio in the Brooklyn engine was about 4 to 1. In regard to the last question, more experiments are needed before it can be answered. Perhaps Mr. Rice can give us some more information on this point as he is the one who decided to use the 3 to 1 ratio.

Mr. Rice.—In adopting this ratio we adopted the foreign practice; and I presume that ratio was adopted abroad for two reasons: First, that it corresponds closely to their saturated steam

practice; and, secondly, by having a large, high-pressure cylinder they are able to use more superheat in that cylinder.

Mr. A. E. Cluett.—I would like to inquire whether outside of the cylinder-ratios any data is available regarding the amount of superheat that could be applied to an ordinary compound engine of about the same size as the one under discussion, without endangering the engine in any way?

Mr. Rice.—I cannot answer this, because it depends largely on the character of the load, the point of cut-off in the cylinder, and various other elements; but in general a temperature in existing engines of over 500 degrees Fahrenheit would give trouble.

Prof. W. F. M. Goss.—It has seemed to me that the use of superheated steam in connection with multi-cylinder engines is somewhat illogical, and I am interested to know whether Professor Jacobus regards the results of his tests as a clear justification for such a combination. The large amount of cylinder-condensation, and the consequent low efficiency which attend the action of the single-cylinder engine may be reduced in several different ways. One way is to multiply the number of the engine cylinders, and another way is to preserve the engine in its simplest form and employ a superheater. Either plan tends to the same result. In either case, also, the improvement is at the cost of added complication in the design of the plant. In one case, this complication takes the form of added cylinders, in the other of an independent device in the form of a superheater. My point is that by following either one of these lines of development to the exclusion of the other, results can easily be obtained which approach closely those given by the compound superheating plant described by Professor Jacobus. If this is true, where is the argument in favor of adding to a plant complications of two different sorts. If, for example, a compound engine will not give the desired degree of efficiency, would it not in most cases be better practice to endeavor to better the performance by extending the series to a triple expansion type, rather than by introducing a device requiring so much attention as a superheater? Or, if it has been decided to employ superheated steam and to meet the problems incident to such practice, then why not seek the full advantage of its use by reducing the engine to its simplest form, using a single cylinder only.

I do not wish to be understood as criticising in the least the

excellent paper which Professor Jacobus has presented. I merely raise the question as to whether the combination which is presented in the engine he describes is one which is likely to find favor in future practice. In view of his very careful study of the subject, a statement from Professor Jacobus as to the arguments which go to sustain the peculiar combination of elements which the engine he describes presents would be most valuable.

*Professor Jacobus.**—In reply to Professor Goss will say that I feel that the gain in economy secured by superheated steam may in many cases justify its use. It is true that with saturated steam we may step from the simple engine to the compound, and from the compound to the triple with a gain at each step, but it also appears that in any type of engine there may be a gain due to superheating. The main point to be determined is whether the additional gain in economy due to the superheating will more than offset the interest on the increased capital investment, and the item for depreciation and repairs of the superheater. Mr. Rice has pointed out that if a compound engine using saturated steam is proportioned to give the best economy the mean effective pressure would be so low as to lead to large dimensions, and that when so constructed its cost is comparable to that of a superheated steam engine and its superheater complete. One thing is certain, and that is that the advisability of using superheated steam depends on the cost of the coal used. If coal is expensive it may pay to use superheated steam, whereas with a cheaper coal it may not. I hope to go into this matter of costs more carefully and presents the results at a later meeting.

Mr. Geo. H. Barrus kindly pointed out after reading my paper that by basing the heat consumption on the temperature at which the superheated steam is furnished to the engine, I have neglected any difference that there may be with saturated and superheated steam in the radiation of the steam mains leading from the boiler to the engine. Any excess of radiation with the highly superheated steam over that for saturated steam, should be charged against the superheated steam plant and the saving that is shown for the superheated steam should, therefore, be somewhat less than I have given. Taking the Millbourne Mills engine there would be a difference in the radiation of the steam mains with

* **Author's Closure under the Rules.**

saturated and superheated steam, which would affect the economy by about one per cent. The steam main of this engine is, however, excessively long, being over twice the length that might ordinarily be used. It is fair to say, therefore, that where the steam main is of the ordinary length and is covered in the same way for saturated and superheated steam, the results for the heat consumption with superheated steam, computed as has already been explained, should be discounted by about one-half of one per cent.; a quantity which is well within the range of accuracy of the tests. As the covering for superheated steam is usually heavier than for saturated steam, this small difference due to the radiation of the steam main will be still further diminished. The mains carrying superheated steam at the Millbourne Mills are 6 inches in diameter, and are covered with a double layer of asbestos and magnesia covering. The inner layer is asbestos fire felt 1⅛ inches in thickness, and the outer magnesia strips of about the same thickness held in place by wire and plastered with asbestos cement. The average outside diameter of the covering is 12½ inches.

No. 1025.*

STANDARD UNIT OF REFRIGERATION.

BY J. C. BERTSCH, ATLANTA, GA.
(Member of the Society.)

1. Since three-quarters of a century ago, mechanical refrigeration has been in practical use. For the first period of about fifty years it was applied most exclusively for making ice and cooling liquids in small quantities only. But during the past twenty-five years its applications have increased so wonderfully and on such a large scale that it must now be taken care of as a special line of business by the refrigerating engineer, and it can no longer be handled in connection with any other branch of mechanical engineering.

2. Refrigeration is now successfully applied in numerous industries, either as a valuable assistance or as a most necessary part for obtaining quicker and better results in the manufacture of many articles.

3. As an exclusive process mechanical refrigeration is used for making ice, preserving perishable goods, controlling fermentation, fractional distillation, assisting in mining and for many other purposes. It will sooner or later be applied to our dwellings and public halls, not only to replace the ice in the refrigerators, but to furnish also cooler and better air during the hot season of the year.

4. In spite of the many years of practical application, and in spite of the thousands of refrigerating machines in operation all over the civilized world, no standard unit of refrigeration has as yet been established. We have to-day no commonly adopted proportions or uniform rules to measure the actual work produced by the refrigerating machine save the adoption of 284,000 British thermal units as an equal to the work accomplished by melting one

* Presented at the New York meeting, December, 1903, of the American Society of Mechanical Engineers, and forming part of Volume XXV. of the *Transactions*.

ton of 2,000 pounds of ice of 32 degrees Fahr. to water of 32 degrees Fahr. But even this value is not properly applied, inasmuch as it is correctly used for calculating the work equal to ice-melting only; as soon as ice-making is to be calculated, two tons of ice-melting are considered as equal to one ton of ice-making, regardless of the fact that under ordinary condition the making of one ton of ice amounts only to about 410,000 to 420,000 British thermal units instead of 568,000 British thermal units.

5. The absence of a standard unit, proportions or rules is the cause of much confusion and many disputes. Everyone is at liberty to choose any rating he sees fit, and as each rating or guarantee can be substantiated under certain assumptions, the purchaser of a machine is at a loss to know what he is getting for his money.

6. We find to-day the capacities of machines based on a condensing water of from 50 up to 75 degrees; compressor displacements of from 6,500 up to 8,500 cubic inches per ton-minute, and guarantees for power required of from one up to two horse-power per ton refrigeration in 24 hours.

7. Realizing that such conditions can not be tolerated much longer, and that uniform rules for this special branch of engineering are badly needed, the Southern Ice Exchange, in convention at Atlanta, Ga., last February, took the lead in appointing a committee for standardizing the proportions of the machinery and apparatus needed for ice-making and refrigeration.

8. Some other and similar associations did likewise, and the Ice Machine Builders' Association took up this matter also, in order to assist in the establishing of uniform regulations for the benefit of all concerned.

9. Guided by the high standing this Society holds throughout the engineering world, I desire to invite your hearty coöperation on this important subject, hence this paper.

Ammonia Compression System.

10. The overwhelming majority of refrigerating machines in actual operation are of the ammonia compression system, which should therefore be considered first. Afterwards the ammonia absorption system, the carbonic acid compression system, vacuum air system and others could be considered along the same lines.

11. With the ammonia compression system pure anhydrous liquid ammonia is evaporated under a low pressure within a system

of pipes. Such a vaporization is only possible if heat is supplied and such heat is furnished by the substance—air, water, brine, etc.—surrounding said pipes, whereby consequently the substance becomes refrigerated and the ammonia takes the form of vapor. The office of the refrigerating machine is to compress this vapor and to discharge the compressed gas into the condenser, where it is condensed—liquified—for further use in the refrigerating system.

Properties of Ammonia.

12. The properties of the ammonia in the state of liquid and vapor has been calculated by Professor De Volson Wood. They are well known and universally accepted as correct.

13. We know exactly what the ammonia can do under certain conditions, and as the temperature, pressure, volume and weight of same are of fixed relations to each other, we can calculate two of these properties as soon as two of them are known.

Standard Ton of Refrigeration.

14. The quantity of refrigeration is now expressed in tons of ice-melting.

As the latent heat of ice is 142 British thermal units, it takes for melting or making one ton of 2,000 pounds of ice of 32 degrees the quantity of 2,000 × 142, or 284,000 British thermal units.

The correct and better expression which would answer for all purposes would, therefore, be:

"One ton of refrigeration is the latent heat absorbed or set free by melting or making one ton of ice of 32 degrees to or from water of 32 degrees Fahr."

Quantity of Ammonia Evaporated.

15. To transfer the heat of 284,000 British thermal units by means of mechanical refrigeration a certain amount of liquid ammonia must be evaporated, and to arrive at the actual amount of ammonia needed we must consider the unavoidable losses of heat during the process of vaporization.

16. *First.* The liquid ammonia coming from the condenser has

a much higher temperature than the one at which it evaporates. The difference between these temperatures is a loss which must be deducted from the heat of vaporization due to the boiling point of the ammonia in the refrigerator.

Second. The vapor leaving the refrigerator becomes heated on its way to the compressor, and especially by coming in contact with the walls and parts of the compressor, which are heated to a certain extent during the compression period. This heating of the vapor causes expansion, and consequently a reduction of the weight due to its pressure.

Third. It is impossible to build a compressor which discharges every particle of the compressed gas. Whatever is left in the cylinder at the moment the piston begins the suction period will reëxpand and fill up a certain space, thus reducing the volume of the entering vapor, or, in other words, the displacement of the compressor.

Fourth. More or less loss is caused by leaks through valves and piston, radiation of the different parts of the system, imperfection of the insulation and pipe covering and impurities in the ammonia itself.

17. The first loss is an exactly known quantity, and is, therefore, accounted for separately in all calculations. But all the other losses can not be measured correctly and independently from each other, wherefore they are accounted for in form of a certain per cent. of efficiency of the compressor.

Efficiency of Compressor.

18. Prof. J. E. Denton stated in *Transactions A. S. M. E.*, vol. xii.: " . . . An important deduction from the measurements by meter of the quantity of ammonia circulated, compared with the weight of ammonia accounted for by the displacements of the compressors, is that the latter falls 25 per cent. short. . . ." Such a shortage of 25 per cent., or an efficiency of the compressor of 75 per cent., has not as yet been proven incorrect, and as actual practice agrees indeed closely with such an efficiency, it can safely be considered as correct.

19. Having now the quantity of heat equal to one ton of refrigeration, and also the loss of heat and the efficiency of the compressor as undisputed facts, it would seem a very simple matter to find the quantity of ammonia which must be actually evaporated

under certain conditions in order to produce one ton of refrigeration.

20. But just the choice of these certain conditions upon which the standard unit shall be based has so far prevented an agreement between the parties concerned.

21. For the establishment of a standard unit of refrigeration it is necessary to agree on the speed of the machine and on the temperatures or pressures which shall form the bases for all calculations. As soon as such bases are found and accepted, the entire question is settled.

Speed of Machines.

22. Up to the present time the most serious trouble is the great difference in the speed of the machines.

Some of the builders use good practice and base their machines on a piston speed of from 125 feet for the smallest up to 300 feet for the largest machines.

23. Others are not so particular, and use a speed of from 150 to 450 feet per minute. Still others don't care at all for any proportions, but simply increase the speed of one and the same machine if a larger capacity is needed, just to meet competition regardless of the efficiency or the life of the machine.

24. Such conditions are not known in any other branch of mechanical engineering. Standard regulations like those accepted in steam and electrical engineering should also be possible in refrigerating engineering.

25. Recently it has been proposed to adopt 50 revolutions per minute as the speed for all machines regardless of size. Such an effort is arbitrary and against the principles of modern engineering. It would mean for the refrigerating machines as much as a proposition of 80 revolutions per minute for all steam engines or 500 revolutions per minute for all dynamos.

26. The speed must necessarily be expressed in feet piston travel per minute, and the number of feet must be chosen with due regards to the fact that all compressors have poppet valves instead of positively mechanically governed valves like the steam engine.

For this reason the speed of a compressor can never be as high as the speed adopted for the steam engine.

27. I have been in favor of a speed of 250 feet as a standard. But after a careful comparison of the speed of well proportioned

machines in actual practice, I think it will be necessary to divide the speed in several classes, in order to prevent too great a departure from the present common practice.

28. All machines with a stroke of 8 inches and less are of little or no importance and could be excluded. The machines above 8-inch strokes could be divided into four classes, for which the standard piston speed per minute should not exceed:

180 feet for machines with a stroke up to and including 12 inches;
240 " " " " " " over 12 and including 24 inches;
300 " " " " " " " 24 " " 36 "
360 " " " " " " " 36 inches.

29. A comparison of this proposition with the present rating of the manufacturers of two of the leading machines, shows the following table of speed:

TABLE OF SPEED.		FRICK—PENNEY, Dry Compression Single-acting Machine.									WOLF—LINDE, Wet Compression Double-acting Machine.								
Builder's Rating.	Capacity tons Refrigeration, 24 hours	10	15	20	40	60	80	100	200	350	6	15	25	40	65	85	120	175	225
	Revolutions per minute	90	85	90	70	65	60	60	55	50	70	70	70	70	63	57	57	50	45
	Piston speed, feet per minute	150	170	225	233	260	280	320	330	400	147	175	198	248	263	285	285	300	360
	Stroke, inches	10	12	15	20	24	28	32	36	48	12⅝	15	17	21¼	25	30	30	36	48
Proposed Standard.	Piston speed, feet per minute	180	180	240	240	240	300	300	300	360	180	240	240	240	300	300	300	300	360
	Revolutions per minute	108	90	96	72	60	64	56	50	45	86	96	84	68	72	60	60	50	45
	Capacity tons Refrigeration	12	15.8	21.3	41	55	85	93.3	182	315	7	20	30	39	74	89	126	175	225

Supposing the Displacement of Compressors to be correct.

Temperature vs. Pressure.

A glance at any ammonia table must convince us that the temperature and not the pressure is the basis for all calculations of the capacity of a refrigerating machine. This fact is not at all a matter of choice, but it is dictated by natural conditions.

The change in the temperature of the atmosphere and of the water supply is the cause of changes of the ammonia pressure, but a change in the latter can never produce a change in the former.

All the properties of ammonia are based on the absolute temperature from which the different pressures are derived.

And yet, in spite of this fact, most all calculations of the capacity of refrigerating machines are based on the suction and condensing pressures, which are often expressed in fractions of one-hundredths of a pound, which of course can not be read on any gauge.

The consequence of such a faulty method is that the exact knowledge of the relations between temperature and pressure are seldom fully understood by those who operate this class of machines, and that it is possible that for a place having a condensing water with 85 degrees a machine is furnished of which the capacity is based on a temperature of 60 degrees.

The present method of allowing two and even more tons of refrigeration for one ton of ice-making capacity is also caused by the disregard of temperatures, because the boiling point of the ammonia for ice-making must be much lower than the one taken at present as a basis for refrigeration. I know of a case where a machine of 150 tons refrigerating capacity had to be worked up to its full capacity to produce 65 tons of ice.

The ammonia tables in practical use give for each degree from minus 40 up to plus 100 the absolute and gauge pressures, volumes and weights of the ammonia, but for only two even numbers of pounds gauge pressure (76 and 87 pounds) the corresponding temperatures, and all the others must be calculated or assumed.

This evil can be easily corrected by basing all calculations of the capacity on the

Temperature to be produced, and

Actual temperature of the condensing water.

Temperature to be Produced.

From zero required for sharp freezers up to 60 degrees for certain rooms, the range in temperatures is a very large one.

The standard should be a fair average of the temperatures most needed, and this is no doubt the temperature required for ice-making, combinations of ice-making and cold storage, and refrigerating plants with brine circulation.

A temperature of 15 degrees Fahr. as a basis would therefore be most suitable for these purposes, and would give a boiling point of zero and a suction pressure of about 15 pounds. It could be applied for about 90 per cent. of all plants, and the balance of 10 per cent. for which temperatures below or above 15 degrees are required could be based on especially prepared equations or tables.

Actual Temperature of Condensing Water.

Different parts of the country furnish water of a great difference in temperature. Water as low as 50 is found at some places in the North; an average of 65 degrees in the East; and from 75 to 85 degrees in the West and South. But comparatively few plants can depend on an abundance of water, which is clearly demonstrated by the rapidly increasing demand of cooling towers.

The temperature of water of the greatest part of this country is from 72 to 78 degrees, and wherever a cooling tower is used it will furnish a water of about the same temperature, whether the initial temperature was 60 or 85 degrees, some extreme conditions not counted.

A condensing water of 75 degrees as a basis would certainly be in line with the great majority of actual conditions. A few degrees more or less than 75 could not change conditions very materially, as a difference of 5 degrees does not amount to more than about 1 per cent. more or less capacity, which is small compared with a factor of 75 per cent. as the basis for the efficiency of the compressor.

With a condensing water of 75 degrees and a modern condensing apparatus, or with a special liquid cooler in connection with an old style condenser, a liquid ammonia of 80 degrees can be easily obtained.

Displacement of the Compressor.

The displacement of the compressor is dependent upon the conditions already mentioned. Taking as a standard basis 284,000 British thermal units as the heat value of one-ton refrigeration; 15 degrees as the temperature to be produced; zero as the boiling point of the ammonia to be evaporated; 555,500 British thermal units as the heat of vaporization of one pound at said boiling point, and 0.1107 pound as the weight of one cubic foot of vapor at said boiling point; 80 degrees as the temperature of the liquid ammonia, and 75 per cent. as the efficiency of the compressor, the standard displacement per minute per ton of refrigeration in 24 hours.

$$\frac{284,000}{(555.50 - 80) \times 0.1107 \times 1440 \times 0.75)} = 4.992$$

or 5 *cubic feet* (8,640 cubic inches).

Standard Unit of Refrigeration.

Recapitulating my propositions, the requirements for a standard unit of refrigeration would be as follows:

1. 284,000 British thermal units as the latent heat of 2,000 pounds of ice constitute one ton of refrigeration.
2. The efficiency of the ammonia compressor is 75 per cent. of the theoretical capacity.
3. The limit of piston travel in feet per minute shall be

180 feet	for	strokes	up to and	including 12 inches ;			
240	"	"	"	over 12	"	"	24 "
300	"	"	"	" 24	"	"	36 "
360	"	"	"	" 36 inches.			

4. The temperature to be produced 15 degrees Fahr., and the boiling point of the evaporating ammonia is zero.
5. The temperature of the condensing water is taken at 75 degrees Fahr., and the temperature of the liquid ammonia at 80 degrees.
6. The displacement of the compressor must be 5 cubic feet, or 8,640 cubic inches, per minute for each ton of refrigeration in 24 hours.

Hence we have

$$\frac{0.75 \times 1{,}440 \times (555.50 - 80) \times 0.1107 \times 5}{284{,}000} = 1.0008 \text{ ton},$$

representing actually *one practical ton of refrigeration.*

Ice-Making Capacity.

It may be interesting to find the relations between one ton of refrigeration and one ton of ice-making under the very same condition as taken for the standard unit of refrigeration.

With a cooling water of 75 degrees, the water coming from the re-boiler with 212 degrees must be cooled before it can enter the ice-making apparatus, and it can therefore be taken again at a temperature of 80 degrees. By means of evaporating ammonia it must be cooled to 32 degrees, and after being frozen the ice must be cooled down to 15 degrees. To simplify matters, we will take the specific heat of the ice also at 1 instead of 0.5, and the work to be done would then be the transfer of the sensible heat of $2{,}000 \times (80 - 15) = 130{,}000$ British thermal units, in addition to the latent heat of 284,000 British thermal units.

The total of 414,000 British thermal units would then constitute the heat value of one ton of ice-making, and one practical ton of refrigeration would be equal to

$$\frac{0.75 \times 1{,}440 \times (555.5 - 80) \times 0.1107 \times 5}{414{,}000} = 0.686$$

or 0.68 *ton of actual ice-making capacity.*

The following Table of Capacities give a comparison of the builders' rating with the actual capacities according to my proposition for a standard unit.

It proves conclusively that the machines mentioned are not based on uniform rules. While the capacity of most of the single-acting machines comes very near within the results from applying my proposition for a standard unit, the capacities of most of the double-acting machines show very great differences.

The true proportions between refrigerating and ice-making capacity hold good for the single-acting machines, but not for the double-acting machines, which for several good reasons fall short in actual practice, which fact is plainly demonstrated by the builders' rating for ice-making capacity.

TABLE OF CAPACITY.			FRICK—PENNEY, Dry Compression Single-acting Machine									WOLF—LINDE, Wet Compression Double-acting Machine.									
Builder's Rating.	Capacity	Tons ice-making....	6	8½	11¼	27	38	46	60	130	200	2.4	8	12	20	30	40	75	Not stated.	
		Tons refrigeration..	10	15	20	40	60	80	100	200	350	6	18	25	40	65	85	150	175	225
	Displacement, cubic inches....		7,688	7,716	8,587	7,910	7,441	7,421	8,210	7,873	7,851	7,983	7,421	7,130	7,068	8,144	7,588	7,161	7,124	6,649
	Feet piston speed..............		150	170	225	233	260	280	320	330	400	147	175	194	248	263	285	285	300	360
	Revolutions per minute........		90	85	90	70	65	60	60	55	50	70	70	70	70	63	57	57	50	45
Proposed Standard.	Compressor	Diameter, inches.	7⅞	8¼	9	12	13½	15	16¼	22½	27	5½	9	9½	11	13¼	15¼	20	21	21
		Stroke, inches...	10	12	15	20	24	28	32	36	48	12¾	15	16¾	21¼	25	30	30	36	48
	Revolutions per minute........		108	90	96	72	60	64	56	50	45	86	96	86¼	68	72	60	60	50	45
	Feet piston speed..............		180	180	240	240	240	300	300	300	360	180	240	240	240	300	300	300	300	360
	Displacement, cubic inches....		5 cubic feet or 8,640 cubic inches.									5 cubic feet or 8,640 cubic inches.								
	Capacity	Tons ice-making....	7	9	14	25	32	40	60	112	194	4	12	17	21	41	53	85	100	117
		Tons refrigeration..	10.6	14.2	21.2	37.6	47.7	73.3	88.7	165.6	286.2	6.8	19.1	25.5	31.8	61.8	78.5	125	149	173

Conclusion.

It has been my endeavor to show that the pressures as a basis for calculation are in place and necessary for computing the power required, but that for computing the capacities of refrigerating machines the temperatures are the only correct basis.

As this subject is of great interest to a great number of industries, I beg leave to suggest that a committee may be appointed consisting of members of this Society actually engaged in refrigerating engineering, and that such committee may have the power to coöperate with all the committees already appointed by the associations of builders and owners of ice and refrigerating machinery for the sole purpose to establish a Standard Unit of Refrigeration.

DISCUSSION.

Mr. Gardner T. Voorhees.—In reply to Mr. Bertsch's paper on the Standard Unit of Refrigeration I wish first to briefly review the efforts that have been made in this line.

In my work as a Refrigerating Engineer, for a long time I have been greatly impressed by the great differences of opinion regarding the capacity of refrigerating machines.

In a series of articles that I published in *Ice and Refrigeration* of May, June, July, August, September and October, 1902, entitled "Analyzing the Compressor," I endeavored among other things to indicate a basis for accurately determining the capacity of such refrigerating machines.

My correspondence with the prominent builders of such machines at that time so impressed me with the great diversity of ratings for capacity that I determined to try and have a standard unit ton for refrigerating machines adopted by those most interested.

It was my intention to have personally presented this subject before the Society at its last winter meeting, but my duties in connection with the St. Louis World's Fair Refrigeration Exhibit at that time prevented my being at the meeting as these same duties have at present prevented my preparing a more elaborate paper.

I brought the question to the attention of the Society in letters of December 3 and 22, 1902, and asked that a committee be ap-

pointed to consider the question. In my letter of December 22d, I suggested the following as a basis on which to attempt to frame rules for the standard unit ton for all classes of refrigerating machines.

First: As referred to compression machines, the use of the factor (cubic displacement of compressor per revolution per ton of refrigeration).

Second: For absorption machines, cubic displacement of the liquor pump; per stroke, per ton of refrigeration, with accompanying difference in percentages of ammonia contained in the strong and weak liquors.

I personally brought up the question at the last February meeting of the Southern Ice Exchange at Atlanta, Ga., with the result that I had the honor to be appointed chairman of a committee of five from that association to consider the subject and report.

Through the active efforts of Mr. J. F. Nickerson, the publisher of *Ice and Refrigeration*, and Mr. J. C. Atwood, manager of the National Ammonia Co., similar committees were appointed at the last March meetings of the Southwestern Ice Manufacturers' Association at Dallas, Texas; The Indiana Ice Manufacturers' Association at Indianapolis, Ind., and the Western Ice Manufacturers' Association at Kansas City, Kansas.

Since the above committees were appointed the Ice Machine Builders of this country have organized an association and have devoted some little time to discussion and tests that may help to establish such a unit.

Numerous articles bearing on the subject have been published from time to time in the columns of *Ice and Refrigeration* by Prof. J. E. Siebel; Eugene T. Skinkle, J. C. Bertsch, A. E. Siebert and myself. For the benefit of those who may be interested to review those articles I give the following references to the pages of *Ice and Refrigeration:*

1903.

Month.	Page.	Author.	Subject.
March...	88d–88e...	G. T. Voorhees ...	Introduction of subject at Southern Ice Exchange Meeting.
May ...	194 ...	Ice Machine Builders...	Proposed rule for Standard Unit Ton.
June ...	232 ...	" " " ...	Criticism of Builders Unit by editorial.
July ...	16–18 ...	E. T. Skinkle ...	Capacity of Compressor, with tables.

1903.—(Continued.)

Month.	Page.	Author.	Subject.
Aug.	... 47–50	... G. T. Voorhees	... General review of subject. Rules suggested and curves for capacity. Criticism of Skinkle's tables.
Aug.	... 50–51	... A. Siebert	... Criticism of Skinkle's tables and proposed new formula.
Sept.	... 110	... G. T. Voorhees	... Diagramatic representation of ammonia in a compression cycle. Criticism of Siebert's formula.
"	... 101-102	... E. T. Skinkle	... Reply to criticisms of Voorhees and Siebert.
"	... 102–103	... Prof. J. E. Siebel	... Criticism of Siebert's formula.
"	... 103	... A. Siebert	... Criticism of Voorhees' articles.
Oct.	... 129-131	... G. T. Voorhees	... Reply to Skinkle's criticism with curves showing Voorhees', Skinkle's and Siebert's results and correction to apply to page 110 of September article.
"	... 125–128	... J. C. Bertsch	... General discussion of subject, with tables, together with discussion of Voorhees', Skinkle's and Siebert's articles.
"	... 131-132	... A. Siebert	... Reply to criticism of Siebel, Voorhees and Skinkle.
"	... 140	... Ice Machine Builders	... Brief notice of tests for unit ton.
Nov.	... 195–196	... G. T. Voorhees	... Criticism of Bertsch's article.

A brief summary of the points discussed are as follows:—Superheating effect, formulæ for capacity, rules for capacity, general curves and tables.

The rules proposed by me in these articles were as follows:—

For Ammonia Compression Machines.

First: A standard unit ton of refrigeration is 284,000 British thermal units.

Second: A refrigerating machine must operate continuously for 24 hours to do refrigeration equal to its rated capacity.

Third: A refrigerating machine shall operate at fifty revolutions per minute when rated at its standard capacity.

Fourth: The standard displacement of an ammonia compressor of one ton capacity is five cubic feet per minute.

Fifth: The approximate displacement per minute of a compressor per ton of refrigeration at a back pressure other than 15

pounds per square inch gauge and a temperature of water to the condenser other than 75 degrees Fahr. is 5 cubic feet plus or minus .18 cubic feet for each pound that the back pressure is below or above 15 pounds and plus or minus .001 cubic feet for each degree that the temperature of water to the condenser is above or below 75 degrees Fahr.

Referring now to Mr. Bertsch's valuable paper, I offer the following criticisms:

The statement, article 14, "One ton of refrigeration is the latent heat absorbed or set free by melting or making one ton of ice of 32 degrees to or from water of 32 degrees Fahr." would seem to be more concise if stated as follows:

One ton of refrigeration is the measure of the heat taken up or given out by melting or making 2,000 pounds of ice of 32 degrees Fahr. to or from water of 32 degrees Fahr. This removes the objection of synonymously using (ton of refrigeration) and (latent heat).

"Given out" seems to be more appropriate than "set free" and "2,000 pounds" obviates the question as to whether a short or long ton is meant.

For the latter part of article 15, the phrase: "and to arrive at the actual amount of ammonia needed, we must consider the unavoidable losses of heat during the process of vaporization," seems more correct if it reads: "and to arrive at the actual amount of ammonia needed we must consider the unavoidable *gain or loss of heat to or from the system during the process of refrigeration.*"

In the second clause of article 16, only the effect of a dry compressor is noted; whereas a wet compressor would have additional weight of ammonia introduced through the expansion valve to counteract the superheating effect during compression.

Article 19, after speaking of the 25 per cent. loss in capacity of compressor in article 18 the statement that the efficiency of the compressor is an undisputed fact seems erroneous in two respects.

First: That the use of the word efficiency is not correct. It might be worded as follows, "and also the loss of heat and the *effective capacity* of the compressor as undisputed facts."

Second: That the effective capacity of the compressor is a very much disputed fact. I admit that the work and tests of Professor Denton are the best data we seem to have at the present time,

and that probably the 25 per cent. loss of capacity is *correct for certain back and condenser pressures.* But I firmly believe that future experiments will show that the effective capacity of the compressor is a variable quantity depending on the back and condenser pressures, the nature of the compressor (wet or dry) and the area and nature of the surfaces that the refrigerant comes in contact with between the time that it leaves the cooler until it is trapped in the compressor.

I believe that the loss of capacity will be more than 25 per cent. for back pressures less than 15 pounds gauge and condensing water hotter than 75 degrees Fahr. and less than 25 per cent. for back pressures more than 15 pounds gauge and condensing water colder than 75 degrees Fahr.

My theory closely follows the cylinder condensation phenomena of the steam engine and seems to be borne out by general experimental data.

Articles 25, 26, 27 criticise my suggestion for the adoption of a standard number of revolutions for the compressor and suggest a sort of sliding scale of piston speeds.

I believe the general rules to be adopted should be as simple as possible, and I suggest that if a standard number of revolutions can be agreed upon that it will much simplify the final rules. For instance, the variation in piston speed suggested by Mr. Bertsch is 100 per cent., while from an inspection of the tables published in his paper on page 7 it is evident that there is not such a large variation in the number of revolutions.

It is not necessary to maintain the arbitrary proportion between the diameter of piston and length of stroke as in these tables. It is evident that where the length of the stroke is twice the diameter of the cylinder (as is the case with many up-to-date machines) that a fixed number of revolutions as 50 would give just the same displacement as a greater number of revolutions and a smaller stroke for the same diameter of piston.

Why not, as new patterns are gotten out, make the smaller machines with extra long strokes and the larger machines with shorter strokes so that a standard number of revolutions may be adopted?

My argument as advanced in favor or revolutions in place of piston speed is as follows, from page 49, August, 1903, *Ice and Refrigeration.*

"Why should not a compressor run at a high piston speed like

a steam engine? The reason is that the valves of a steam engine are positively mechanically governed and must act positively at each revolution of the engine. A compressor usually has poppet valves which are not positively mechanically governed, but depend for their action upon a difference of pressure of a gas on the opposite faces of the valves. Such valves have an appreciable inertia and can only reciprocate a limited number of times per minute without requiring an excessive difference of pressure on the two faces of the valve. Such a difference of pressure is out of the question where low back pressure is to be maintained in the cylinder during the suction stroke.

"A too rapid reciprocation of the piston will either cause the suction valve to so act that the pressure in the cylinder is much less than that in the suction pipe, or else the rapid motion of the piston will not give the valve time enough to overcome its inertia and change the direction of its motion, with the result that the valve will never be fully open or fully shut. The result of such action evidently greatly cuts down the capacity of the compressor so that it would not do as good work at quite high speeds as it would at slower speeds.

"We all know from the steam engine that the limit of possible piston speed in the compressor is never reached so far as operating the piston and stuffing box are concerned. It seems to me that the piston speed should not enter into the question. The vital question seems to me to be: How many times per minute can the valves be made to reciprocate to advantage? If I owned and operated a compressor I would, from my experience, set the limit at fifty revolutions per minute.

"Whatever the number of revolutions finally settled upon as being advisable it is evident that it will apply equally well to a large or a small machine so long as the machine has equal valve area for equal weights of gas pumped at equal pressures. A consideration of this may show a marked difference not only in the different makes of machines but in different sizes of the same makes of machines; and may lead in the future to different proportions between diameter and stroke so as to give larger valve areas."

In addition to the above I will repeat a remark that was made to me by a very successful refrigerating machine builder, "I find it cheaper to install ample compressor displacement rather than to install extra expansion coils, condensers, ice cans, etc."

There is a whole volume of common sense in the above statement and a careful study of its results would, I think, lead to a conclusion that a slow running machine is a good investment both for the builder and the purchaser.

Mr. Bertsch implies that we should adopt temperatures rather than pressures because temperatures are given in even numbers in the ammonia tables while pressures are usually expressed as fractions. This seems to be no argument, for, if desired, ammonia tables can easily be calculated with pressures in whole numbers and temperatures generally in fractions.

Mr. Bertsch assumes that there should be a difference of 15 degrees Fahr. between the boiling point of the refrigerant and the temperature of the substance to be cooled. A difference as great as this is not found in modern practice where shell or double tube brine coolers are used, for with such coolers a minimum difference of temperatures of from 3 degrees Fahr. to 5 degrees Fahr. is usual while 10 degrees Fahr. difference is a maximum.

Referring to the rules, the first one is not well worded, while for rule two, I suggest that the word efficiency should be changed to *effective capacity*. Rule three does not seem to be simple enough. Rule four is too arbitrary and not based on standard conditions to give the best economy; I advise back pressure as a basis to figure on.

In obtaining the factor 0.68 as the multiplier to be applied to the tons refrigerating capacity to obtain the tons ice making capacity, no account has been taken of the gain of heat by radiation to the ice making tanks or of the loss of ice in thawing from the cans or plates.

I wish to call attention to the rule adopted by the Ice Machine Builders' Association as published on page 194 of the May, 1903, number of *Ice and Refrigeration* which reads: "It is acknowledged by the Ice Machine Builders' Association of the United States, here assembled: that, in the operation of refrigerating machinery it requires the evaporation of 27.7 pounds of anhydrous ammonia per hour at a pressure of 15.67 pounds above atmosphere, condensing pressure to be taken at 185 pounds above atmosphere, to produce an effect equal to the melting of one ton of ice per twenty-four hours, and that the capacity ratings of refrigerating machines should be figured on this basis."

This rule seems to be framed much as that for the standard boiler horse-power and it is good as far as it goes, and its form

would be ideal if it were as convenient to measure ammonia as it is to measure water in a boiler test, or if the question of the speed of the machine were considered. Metering anhydrous ammonia is very uncertain and requires specially constructed meters. I have used the same meter on some tests that Professor Denton used on his tests and I found that unless great care was taken in properly installing the meter, much gas would be generated in the liquid pipe and in the meter which would make the meter buzz around like a top with results that were anything but reliable.

Furthermore, it would seem to me to be very questionable as to whether anhydrous ammonia could be successfully measured without using special apparatus and the services of an expert whose services should not be required in so simple a matter.

The Ice Machine Builders have shown great interest in this subject and have started to make tests on one make of machine with the evident purpose of gathering data upon which to base the standard unit ton.

I think it unwise to attempt to derive data that is supposed to stand for all time from a single make of machine, especially if that machine is erected for the sole purpose of making tests. I think data should be obtained from all recognized standard makes of machines and that tests should be made at the places where the machines are in actual commercial operation.

Each machine builder can single out one or more of his best machines and have a careful expert test made by his own experts under the supervision of some reliable and impartial expert.

A general review of these results should give the necessary data for the final action, and then every builder will have had a chance to have his say and to submit the indisputable value of his machine as shown from verified tests.

At first thought I feel sure that many of the machine builders will object to this method as they seem to have a mistaken idea that their competitors would get the better of them and that the public in general would learn too many things that they should not know. However, I believe that a careful consideration will convince most of the builders that such a course would have many advantages in bringing to light the best points of all machines so that all could profit thereby in discarding old and undesirable types and gradually adopting the better and more modern methods.

Let the public know more about ice machines and remove the

general impression that there is something very unusual in their construction and operation and the public will very shortly want more machines.

It is my intention to offer every facility for such tests to be made upon the several different types of machines to be exhibited at the St. Louis World's Fair next year, and I will be glad if all the recognized experts in this or any country will get together and suggest methods for and help conduct such tests.

In closing I wish to apologize for taking so much time and touching on points not covered in Mr. Bertsch's paper.

My only excuse is the great interest I take in this subject.

Mr. Thos. Shipley.—It is because I do not believe that some of the statements made in the paper under discussion should be allowed to go on record as facts, that I felt called upon to make these remarks.

It is the fact that no unit of refrigeration has been adopted by the engineers engaged in the manufacture or operation of ice and refrigerating machinery, and it is also the fact that every engineer so engaged has felt the need of the adoption of some unit upon which the commercial rating of ice and refrigerating machinery could be based.

The unit, if it may be called such, must be based upon an adopted back or suction pressure, as upon this back or suction pressure depends the conditions under which the ammonia is evaporated and the work done.

I speak of ammonia as the refrigerant; it is the one that is almost universally used, and also because to attempt to adopt a separate unit for every refrigerant would be an endless job and not advisable at this state of the art.

When once a standard back pressure has been adopted, then the unit has been arrived at; the back pressure has been the bone of contention and must be agreed upon before anything else can be done.

Mr. Bertsch makes the statement that the efficiency of a compressor is 75 per cent. and bases this statement on deductions made by Professor Denton years ago, and he further states that this efficiency has not been proven incorrect.

In this I must correct Mr. Bertsch; 75 per cent. efficiency is incorrect, and has been proven so conclusively, especially in the recent tests made at York by a committee selected from the

manufacturers of ice and refrigerating machinery. The three members who directed the test were Geo. Richmond, representing the De La Vergne Refrigerating Machine Co.; N. H. Hiller, representing the Carbondale Machine Co., and myself, representing the York Manufacturing Co.

The fourth member, Mr. Theo. Vilter, of the Vilter Manufacturing Co., was unable to be present at the tests.

The three active members of this committee are all members of this Society.

The compressor we used developed an efficiency of 83 per cent. under 15.67 pounds back pressure.

The plant used in making the tests was put up for that purpose, and every means possible was taken to guard against errors.

A mercury column was used to determine the back pressure, as we found gauges were not reliable for the purpose. We also weighed the liquid ammonia, as it was impossible for us to get a meter which would handle liquid ammonia accurately.

The plant was of sufficient size to warrant accuracy, and the tests were run for 6 days consecutively, hence the compressor efficiency obtained can be relied upon.

The compressor was operated at an average of about 70 revolutions per minute.

As to the proper speed at which a compressor should be operated, I will say that from tests which are now being carried on on the same test plant under my direction, it has been shown that the efficiency of a compressor increases as the revolutions increase.

The speed tests were made for each 10 revolutions from 40 to 100, the compressor being 18-inch stroke.

There is no question whatever that the efficiency of the different types of machines varies, hence the adoption of a standard compressor displacement per ton of refrigerating would not be possible any more than it is possible to adopt a standard amount of steam per horse-power for all types of engines.

Prof. S. A. Reeve.—The writer wishes in the first place to commend the enterprise of Mr. Bertsch in bringing this subject before the Society. It is one which has long needed attention.

In the second place, he would suggest the advisability of submitting this matter to the consideration of a committee, for the formulation of the Society's views as to the adoption of some such standard unit. To that end he would suggest that a committee

be appointed by the President, to report to the Society at its next annual meeting.

Thirdly, he would suggest to such committee, if one be appointed, the need for a standard method of expressing the *efficiency* of a refrigerating or ice-making machine as well as for a standard unit of refrigeration; for we have no recognized method now. To start the discussion, he would suggest, tentatively, the following method:

Let Q be the standard unit of refrigeration, supposedly the 284,000 British thermal units suggested by Mr. Bertsch.

Let the lower limit of temperature be the zero-point (F) suggested by Mr. Bertsch or, in general, any absolute temperature T_1 which might be adopted in its place, or which might be left undecided for variation to suit each case.

Let T_2 be the upper limit of temperature, that of the condensing water, either the 536 degrees absolute (75 Fahr.) suggested by Mr. Bertsch, or the particular temperature applying in any given case.

Then the least amount of energy, measured in British thermal units which could possibly accomplish the unit of refrigeration would be, in general,

$$\frac{T_2 - T_1}{T_1} 284.000 = Q_w,$$

or, supposing the adoption of Mr. Bertsch's standard temperatures, $Q_w = 46,204$ British thermal units, or very close to $\frac{3}{4}$ horse-power per ton capacity.

In any given case let the *actual* energy absorbed in producing the standard unit of refrigeration be q_w, which will always be some quantity larger than Q_w. Let the efficiency of any such a case be expressed as

$$F = \frac{q_w}{Q_w}.$$

This method, it is true, takes no cognizance of the fact that not even a perfect ammonia-machine could ever hope to reach 100 per cent. efficiency. For this reason the writer would personally prefer to see the expression for the efficiency of the machine referred to that of a perfect ammonia-machine, which

would always have an efficiency less than that expressed by $(T_2 - T_1) \div T_1$. But because he finds that the average engineer looks askance at and will not adopt a quantity so complex in its computation as the efficiency of the perfect ammonia-compression-machine, he suggests the use of the simpler expression just given.

Mr. S. H. Bunnell.—The adoption of an arbitrary capacity unit is much like certifying officially that all machines have the same efficiency. The condition imposed on the makers of refrigerating machinery is that required of all constructors of machines—that they shall be able to perform what they promise. To do this satisfactorily generally means that the speed of machines must not be too high, the consumption of power too great, or the liability of accident too imminent. If the manufacturer can construct an apparatus of superior efficiency with out sacrificing essentials such as these he should have the benefit of his efforts.

The high acting slow speed American compressor with pistons running to practical contact with heads, must necessarily displace more ammonia per stroke than the double-acting machine of same cylinder volume, but the latter may make up for clearance and valve loss by saving in friction losses on account of avoiding the idle stroke. An arbitrary unit based on cylinder dimensions alone favor small and inefficient machines by giving them a rating higher than they deserve.

The builder designs his compressor with regard to the universal requirement of maximum efficiency at minimum expense, and selects speed, dimensions, style, materials and workmanship in accordance with his judgment. He must not build or run his refrigerating compressor on other than satisfactory compressor lines. But the ratio of useful effect to theoretical capacity depends on his ability as designer and constructor, not on mere cylinder volume. The purchaser of a refractory plant wants useful effect, and generally requires a practical demonstration, with due regard to the power consumption and other expenses. If a constructor wants to furnish cylinders with a larger clearance in order to obtain certain advantages in design of valves, the provision of an arbitrary capacity unit gives him the right to claim more for his machine than it can do in comparison with some single-acting machine without appreciable clearance. Neither 75 per cent. or any other figure can cover all cases correctly.

We have already two useful units of refrigeration; one, that defined by Mr. Bertsch and universally used in America, and the other, the arbitrary ice-making unit, the ton of ice-making capacity, equal to two tons of refrigerating or simple cooling capacity. The ice-making unit is based on practical results; for the ice made is weighed after melting from cans or cutting from plates, and some of the refrigeration is dissipated in leakage through tank walls and other losses besides direct water cooling. The proposed ice-making standard could only be approached in large plants.

A practical and absolute standard has been adopted by the Ice Machine Builders' Association of America. It may be compared with the definition of paragraph 6 of the recapitulation. With a condensing pressure of 185 pounds gauge, the displacement of the compressor must be such that it will pass to the condenser .462 pounds of saturated ammonia vapor evaporated at 15.67 pounds gauge pressure per ton per minute. I do not see the force of the point made against pressure readings and in favor of temperatures. Pressure gauges must be used for safety's sake, and may well be supplemented by thermometers; but since pressures and corresponding temperatures of saturated ammonia are shown side by side on the gauge dials their relation is always apparent. Fractions of pounds or degrees are of no importance in practical design and operation.

Mr. Wm. T. Magruder.—Will the author please inform us why he prefers to use (paragraph 14) "142 British thermal units" as the latent heat of ice, when the more generally accepted figure is 144 British thermal units.

Professor Jacobus.—This subject would be made much clearer if the paper were divided in two parts; first, that relating to standard units of refrigeration; and second, that recommending some standard way of rating the capacity of refrigerating machines. If this were done the first part would be very simple, and the standards now in common use,—the ice-melting capacity and the ice-making capacity,—would probably be all that would need to be considered. When we come to the second part, however, and attempt to specify what shall be the standard rating of a refrigerating machine, we have a complicated problem to deal with. Some classes of machines will do better work relatively to others at high temperatures of refrigeration and others at low; some are handicapped by warm condensing water to a greater

extent than others, and it will be very hard to say what set of conditions will be fair for all machines or whether any set of conditions can be selected which will be fair for all.

Finally, even if a standard set of conditions are selected on which to base the rating of machines, it will be necessary to know the actual conditions under which a machine is to be operated before we can estimate the work that the machine should be able to accomplish.

*Mr. J. C. Bertsch.**—The discussions have furnished some valuable statements, and I am much pleased that this subject has met with such great interest.

Mr. G. T. Voorhees places himself with many of his criticisms in the attitude of a corrector by using hair-splitting methods which should have been omitted. At the present stage of the question it is immaterial whether we say "Absorbed" or "Taken up:" "set free" or "given out;" "efficiency" or "effective capacity," etc.

While I endeavor to fit the Standard Unit as much as possible to the present conditions, Mr. Voorhees desires to change most of the existing conditions to make them fit his proposed rules. If he would study manufacturing methods and machine shop practice, he would surely realize the impossibility to change the existing proportions of the machines so radically; to make new ammonia tables, and so on.

Mr. Voorhees proposed first to consider pressures only in calculating the refrigerating capacity. My article on that subject in *Ice and Refrigeration*, of October, 1903, induced him to "amend" his rules, and he adopted from my proposition the temperature of condensing water for his condensing pressure. Now, if temperatures are the proper thing for one side, why shall they not be proper for the other side of a system?

The fact that differences of from 3 to 5 degrees between the refrigerant and the substance to be refrigerated are sufficient when modern apparatus are used does not justify the making of such a difference the standard. At least 90 per cent. of the existing and future plants are and will be of the "old-time brine tank style," for which the Standard Unit must also fit. Besides, if such small differences should be adopted, then a compressor displacement of 5 cubic feet per ton is much too large, or an efficiency of 75 per cent. is much too high.

* Author's Closure, under the Rules.

A boiling point of zero would produce temperatures of from 3 to 5 degrees, which are far below the practical average. But taking 15 degrees as the actual average temperature to be produced, a boiling point of 10 degrees would have to be assumed, and the displacement of the efficiency would be as follows:

1. *Displacement.*

$$\frac{284{,}000}{(549.35-80) \times 0.1383 \times 1440 \times 0.75} = 4.051 \text{ cubic feet.}$$

2. *Efficiency.*

$$\frac{284{,}000}{(549.35-80) \times 0.1383 \times 1440 \times 5} = 0.6133.$$

Such a displacement of efficiency does not at all agree with actual practice, and is at variance with all, even with Mr. Voorhees's own proposition of a Standard Unit.

Mr. Thos. Shipley has made a full confession in favor of my proposition, while he intended to oppose the same. His principal remarks were:

The efficiency of a compressor is 83 per cent. instead of 75 per cent.

The back pressure has been the bone of contention——

Gauges are not reliable for measuring back pressures——

All of this Mr. Shipley found out by conducting a test with a machine put up for that special purpose, and where "every means possible was taken to guard against errors." This test was made by three experts and for a period of six days.

I beg to ask now every fair-minded engineer to answer the following question: "If a machine, built and put up with all possible care, tested by three eminent experts, with all the assistance to their command regardless of cost, does not give a higher efficiency than 83 per cent., what can be expected of a machine built and put up in competition with ten or more other makes, operated by an ordinary engineer, and oftentimes by some one who knows just how to hold an oil can, run for twenty-four hours per day during a period of about six months, without any appliance to properly control the proper charge of ammonia, running short on condensing water and working against many other unfavorable conditions,—what efficiency can be expected of such a machine representing the average outfit?"

At least 50 per cent. of all the machines in commercial operation will not give an efficiency of 75 per cent., even if they were all of the single acting type. But when it comes to the double

acting machines, with the entire system full of oil, efficiencies of from 50 to 60 per cent. will be a fair average.

If actual tests shall be made for finding a Standard Unit, then let us test machines in commercial operation and under natural and normal conditions, such as the public can afford to maintain. But testing especially built machines at the manufacturer's place by experts and under the most favorable conditions can be called anything but Standard.

Mr. Shipley is in favor of back pressure, but he admits that gauges are not reliable for measuring same. He thinks that upon the back pressure depends the work done.

The logical order of question and answer is simply this: "What work shall be done?" "Producing a temperature of 15 degrees." Well, then we must carry a back pressure of about 15 pounds.

We see on this simple example that the work to be done is the first consideration, and all other conditions, even the back pressure, must be arranged accordingly. The public knows exactly what temperatures are needed, but nobody cares for the pressures carried. Hardly 5 per cent. of all the ammonia gauges are correct for any length of time, which is confirmed by Mr. Shipley's own statement, but the cheapest thermometer does not differ more than one degree. To repair a gauge takes much time and is expensive, because it must be sent to the factory, but a thermometer can be bought in every little town. All the work of a refrigerating machine is controlled by the temperature, and the owner or manager of an ice plant, cold storage house or brewery will certainly not order his engineer to keep certain back pressures, but they simply ask for certain temperatures. These are some of the reasons why I propose to make the temperatures the foundation for a Standard Unit, and not the pressures of the ammonia.

Let the public know what a machine can do and how the work can be controlled, instead of covering defects with rules and terms which can not be understood by many.

The remarks of Mr. S. H. Bunnell are answered by the foregoing with the exception of the item with reference to gauges having the corresponding temperatures marked on the dial. We all know, and Mr. Shipley is the authority for the fact, that ammonia gauges are not reliable, as most of them are from 1 to 15 pounds out. But if the readings of the pressures are incorrect and unreliable, then the corresponding temperatures are also

wrong, and without using a thermometer to ascertain the true state of affairs, nobody knows the conditions under which the work is done.

The question of Professor Wm. T. Magruder I will answer thus: "Since the quantity of 284,000 British thermal units is generally adopted as the cooling effect of one ton, or 2,000 pounds, of ice, the latent heat of ice is accepted as being 142 British thermal units."

No. 1026.*

SPECIFICATIONS FOR BOILER PLATE, RIVET STEEL, STEEL CASTINGS AND STEEL FORGINGS.

Professor Spangler.—You may remember the conditions under which this Committee was appointed, but to make it entirely clear I would like to go into the history of it just a little. There is a society known as the American Society for Testing Materials, which was the outgrowth of the International organization of which we have heard a great deal at meetings of this Society. Committee No. 1 of that Society prepared a series of specifications, and Mr. Webster, at the request of Mr. Hutton, presented these specifications at a meeting of this Society, and asked that a committee be appointed on this particular subject. A committee of five was appointed, consisting of Mr. Cramp, Mr. Kent, Mr. Morison, Mr. Waitt and myself. In the usual way copies of these specifications were sent to various members of the Society, with the usual result—that is, in a few cases, after writing two or three letters, replies were received. The Committee decided to submit, at this time, a report to the Society, subject to revision, asking that the report be sent to all members of the Society, that something like a full written discussion from members who are interested in the subject might be had, and that a revised report be formulated at some future time.

It seems to me to be the proper procedure that, after this Society has finished whatever work it may decide to do, the report, together with the report of all the committees of other societies that may be working on the subject, should go back to Committee No. 1—that is, any report that we might make should be rather an advisory report than an attempt at a finality. This Committee No. 1 is the Committee which will finally, I believe, formulate specifications under which work of this sort is to be done.

With this as an introduction, your Committee would respectfully report as follows:

---

* Presented at the New York meeting (December, 1903) of the American Society of Mechanical Engineers, and forming part of Volume XXV. of the *Transactions*.

SPECIFICATIONS FOR BOILER PLATE, RIVET STEEL, STEEL CASTINGS AND STEEL FORGINGS.

This report is sent out subject to revision, and the Committee asks that written discussion be sent to its chairman that the results may be incorporated in the final report to be presented at the New York meeting of the Society.

The Committee to which was referred the question of specifications for boiler plate, rivet steel, steel castings and steel forgings, reports that it has used the specifications prepared by the American Branch of Committee No. 1 of the International Association for Testing Materials, of which Mr. Wm. R. Webster is Chairman, as the basis of its work, and the changes hereafter noted are recommended in these specifications.

1. That the maximum sulphur in flange or boiler steel be reduced from .05 to .04.

2. That the tensile strength be specified as stated in the table with an allowable variation of 5,000 pounds. That fire box steel be specified at 55,000 pounds instead of 57,000 pounds per square inch. That the determination of the yield point for ordinary grades be omitted.

3. The tensile strength of castings has been modified, the specified value desired being stated, and the variation, 5,000 pounds, being allowed. The values, as recommended by Committee No. 1, and by this Committee, are as follows:—

	Com. No. 1's. Minimum.	Recommended by Committee.
Soft	60,000	60,000 ± 5,000
Medium	70,000	70,000 ± 5,000
Hard	85,000	80,000 ± 5,000

4. The elongation in 8-in. is stated instead of in 2-in. and an increase in elongation of 25% is called for on the 2-in. specimen.

For a 2-in. specimen from castings the corresponding elongations are:

	Com. No. 1.	Recommended by this Committee
Soft	22%	20%
Medium	18%	17.5%
Hard	15%	15%

5. That the 8-in. specimen be made the standard specimen and the 2-in. to be used only when it is inconvenient to use the 8-in.

6. That nickel steel forgings and oil tempered forgings be not included in this specification, because the present state of the art does not warrant general specifications being drawn for these materials.

7. That for soft or low carbon steel forgings the chemical requirements be not over .06 phosphorous, and .05 sulphur, instead of .10 phosphorous and .10 carbon.

8. That for "carbon steel not annealed" the term "medium steel" be used, and that the sulphur be reduced from .06 to .05 per cent.

9. That, wherever it is desirable that the elastic limit be determined, an extensometer be used, and that the elastic limit be taken as "that point at

which the elongation in 8-in. per 1,000 pounds of added stress per square inch first exceeds four ten-thousandths of an inch." *

10. The remainder of the specifications of Committee No. 1 are recommended for adoption, and are here re-arranged.

Standard Specifications for Steel Boiler Plate, Rivets, Castings and Forgings.

Process of Manufacture.

Boiler Plate and Rivet Steel shall be made by the open hearth process.

Castings and Forgings may be made by the open hearth, crucible, or Bessemer process.

Castings may be annealed or unannealed as specified.

Tensile Tests.

Test piece—The standard test specimen shall be eight inches (8") gauged length. The standard shape is shown in Fig. 92.

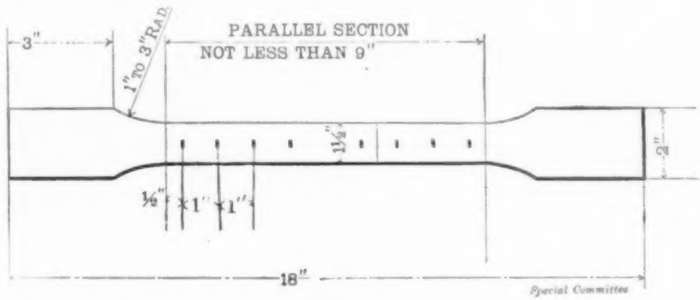


Fig. 92.

Width of specimen along the parallel section shall be 1½ inches, whenever possible.

Thickness of specimen shall be one-half inch or over, whenever possible.

Plates—Two opposite sides shall be the rolled surfaces if not over ¾-inch thick.

Rivets—Rivet rounds and small rolled bars shall be tested full size as rolled.

Castings and Forgings—Specimen may be planed parallel sided or turned parallel for not less than 9 inches in length, the smallest dimension being ½-inch, if possible.

When it is inconvenient to use the standard test specimen the specimen may be made as shown in Fig. 93. In every such specimen the elongation in two inches will be 25% greater than that specified for the standard specimen.

Number of Test Specimens.

If a tensile specimen develops flaws or breaks outside the middle third of its gauged length, another may be substituted.

---

* The "apparent elastic limit," suggested by Prof. J. B. Johnson in his "Materials of Construction," and restated by William Kent in *Transactions of American Institute of Mining Engineers*, 1903.

Plates—One from each plate as it is rolled.

Rivet Rounds.—Two from each melt.

Castings and Forgings—Depending upon the character and importance of the piece.

Location of Test Specimens.

Castings—A test piece shall be cut cold from a coupon to be molded and cast on some portion of one or more castings from each melt or blow, or from the sink-heads (in case heads of sufficient size are used.) The coupon or sink-head must receive the same treatment as the casting or castings, before the specimen is cut out, and before the coupon or sink-head is removed from the casting.

Forgings—The test specimen shall be cut cold from the forging or full-sized prolongation of the same parallel to the axis of the forging and half way between the center and outside, the specimens to be longitudinal, *i.e.*, the length

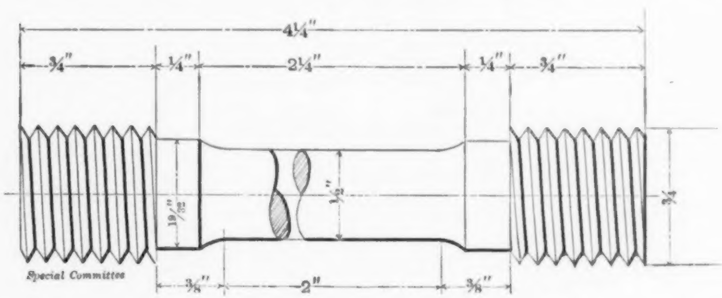


Fig. 93.

of the specimen to correspond with the direction in which the metal is most drawn out or worked. When forgings have large ends or collars, the test specimens shall be taken from a prolongation of the same diameter or section as that of the forging back of the large end or collar. In the case of hollow shafting, either forged or bored, the specimen shall be taken within the finished section prolonged, half way between the inner and outer surface of the wall of the forging.

Bending Tests.

Bending tests may be made either by pressure or by blows.

Cold bending tests are to be made on the material in the condition in which it is to be used. For a quenched bending test the specimen shall be heated to a light cherry-red as seen in the dark, and quenched in water, the temperature of which is between 80° and 90° Fahrenheit.

Test Specimen.

Plates—One and one-half inches wide and if ¾-inch or less in thickness with opposite faces rolled. If over ¾-inch thiek, specimen may be reduced to ½-inch. Edges are to be milled or planed.

Rivet Rounds—Tested full size as rolled.
Castings and Forgings—Specimen one inch by one-half inch.

Number of Test Specimens.

Plates—One cold bending and one quenched bending specimen from each plate as it is rolled.
Rivet Rounds—Two cold bending and two quenched bending specimens for each melt.

Location of Specimen.

Castings and Forgings—As specified for tension specimen.

Chemical Analysis.

Turnings from tensile specimen, drillings from tensile or bending specimen or drillings from small test ingot may be used for chemical analysis.

For locomotive fire box steel check analysis may be required from the tensile specimen of each plate as rolled.

Drop Test.

A test to destruction may be substituted for the tensile test, in the case of small or unimportant castings, by selecting three castings from a lot. This test shall show the material to be ductile and free from injurious defects, and suitable for the purposes intended. A lot shall consist of all castings from the same melt or blow, annealed in the same furnace charge.

Percussion Test.

Large castings are to be suspended and hammered all over. No cracks, flaws, defects, nor weakness shall appear after such treatment.

Homogeneity Test for Fire Box Steel.

A sample taken from a broken tensile test specimen, shall not show any single seam or cavity more than one-fourth inch ($\frac{1}{4}''$) long in either of the three fractures obtained as described below.

A portion of the broken tensile specimen is either nicked with a chisel or grooved on a machine, transversely about a sixteenth of an inch ($\frac{1}{16}''$) deep, in three places about two inches (2″) apart. The first groove should be made on one side, two inches (2″) from the square end of the specimen; the second, two inches (2″) from it on the opposite side; and the third, two inches (2″) from the last, and on the opposite side from it. The test specimen is then put in a vice, with the first groove about a quarter of an inch ($\frac{1}{4}''$) above the jaws, care being taken to hold it firmly. The projecting end of the test specimen is then broken off by means of a hammer, a number of light blows being used, and the bending being away from the groove. The specimen is broken by the other two grooves in the same way. The object of this treatment is to open and render visible to the eye any seams due to failure to weld up, or to foreign interposed matter, or cavities due to gas bubbles in the ingot. After

rupture, one side of each fracture is examined, a pocket lense being used if necessary, and the length of the seams and cavities is determined.

Branding.

Every finished piece of steel plate shall be stamped with the melt number, and each plate, casting or forging and the coupon or test specimen cut from it, shall be stamped with a separate identifying mark or number. Rivet steel may be shipped in bundles securely wired together with the melt number on a metal tag attached.

Variation in Weight.

The variation in cross section or weight of more than 2½ per cent. from that specified will be sufficient cause for rejection, except in the case of sheared plates, which will be covered by the following permissible variations:

Plates 12½ pounds per square foot or heavier, up to 100 inches wide, when ordered to weight, shall not average more than 2½ per cent. variation above or 2½ per cent. below the theoretical weight. When 100 inches wide and over 5 per cent. above or 5 per cent. below the theoretical weight.

Plates under 12½ pounds per square foot, when ordered to weight, shall not average a greater variation than the following:

Up to 75 inches wide, 2½ per cent. above or 2½ per cent. below the theoretical weight. 75 inches wide up to 100 inches wide, 5 per cent. above or 3 per cent. below the theoretical weight. When 100 inches wide and over 10 per cent. above or 3 per cent. below the theoretical weight.

For all plates ordered to gauge, there will be permitted an average excess of weight over that corresponding to the dimensions on the order equal in amount to that specified in the following table:

TABLE OF ALLOWANCES FOR OVERWEIGHT FOR RECTANGULAR PLATES WHEN ORDERED TO GAUGE.

Plates will be considered up to gauge if measuring not over $\frac{1}{100}$-inch less than the ordered gauge.

The weight of 1 cubic inch of rolled steel is assumed to be 0.2833 pound.

Plates ¼-inch and over in thickness.

Thickness of plate. Inch.	Width of Plate. Up to 75 inches. Per cent.	75 to 100 inches. Per cent.	Over 100 inches. Per cent.
¼	10	14	18
$\frac{5}{16}$	8	12	16
⅜	7	10	13
$\frac{7}{16}$	6	8	10
½	5	7	9
$\frac{9}{16}$	4½	6½	8½
⅝	4	6	8
Over ⅝	3½	5	6½

Plates under $\frac{1}{4}$ inch in thickness.

Thickness of plate. Inch.	WIDTH OF PLATE. Up to 50 inches. Per cent.	WIDTH OF PLATE. 50 inches and above. Per cent.
$\frac{1}{8}$ up to $\frac{5}{32}$	10	15
$\frac{5}{32}$ " $\frac{3}{16}$	$8\frac{1}{2}$	$12\frac{1}{2}$
$\frac{3}{16}$ " $\frac{1}{4}$	7	10

Finish.

All material must have workmanlike finish.

Plates must be free from injurious surface defects and laminations.

Castings must be true to pattern, free from blemish, flaws or shrinkage cracks. Bearing surfaces shall be solid and no porosity shall be allowed in positions where the resistance and value of the castings for the purpose intended will be seriously affected thereby.

Forgings must be free from cracks, flaws, seams or other injurious imperfections, and must conform to dimensions.

Inspection.

The inspector representing the purchaser shall have all reasonable facilities afforded to him by the manufacturer to satisfy him that the finished material is furnished in accordance with these specifications. All tests and inspections shall be made at the place of manufacture, prior to shipment.

Respectfully submitted,

H. W. SPANGLER, *Chairman.*

STEEL.	CHEMICAL PROPERTIES. Phosphorus (not over), per cent.	Sulphur (not over), per cent.	Manganese, per cent.	PHYSICAL PROPERTIES. Tensile strength, lbs. per sq. in. (Allowable variation, ± 5,000 lbs.)	Elongation in 8 inches, per cent.	Contraction of area, per cent.	BENDING. Around a diameter of—	Through — degrees.
BOILER PLATE & RIVET:								
Extra soft...	.04	.04	.30 to .50	60,000	28 †	....	Flat.	180
Fire box...	Acid, .04 Basic, .03	.04	.30 to .50	55,000	26 †	....	Flat.	180
Flange or boiler....	Acid, .06 Basic, .04	.04	.30 to .60	60,000	25 †	....	Flat.	180
FORGINGS:								
Soft........	.06	.05	....	60,000	22	35	½″	180
Medium.....	.06	.05	....	70,000	16	30	1½″	180
High........	.04	.04	....	80,000*	18	35	1″	180
CASTINGS. (When physical requirements are not specified carbon must be less than 40 per cent. and phosphorus less than .08 per cent.):								
Soft........	.05	.05	....	60,000	16	30	1″	120
Medium.....	.05	.05	....	70,000	14	25	1″	90
Hard.......	.05	.05	....	80,000	12	20	....	....

* For carbon steel, to be annealed and having no diameter nor thickness greater than 10 inches, allow a reduction of 1,000 pounds for each additional inch in diameter or in thickness of section.

† For material over $\frac{3}{4}$ inch thick deduct 1 per cent. for each $\frac{1}{8}$ inch excess. For material under $\frac{5}{16}$ inch thick deduct $2\frac{1}{2}$ per cent. for each $\frac{1}{16}$ inch decrease.

The Committee submits this as a tentative report, and asks for it the careful consideration of the members of the Society who are interested in the subject.

Mr. Henning.—From what Professor Spangler has said I am simply amazed. There are statements made in the report which cannot be supported. The determination of a very important property indicated by the "yield point" has been dropped because it has become the custom in our mills to run machines at such speed as to make it impossible to determine it. Now, I am going to stand and fight for this, the determination of this point, until I am dead. It is time to put a stop to such preposterous audacity. I tell you, gentlemen, as engineers, that we should rather determine the permanence and the actual strength of all machines and structures, not by the ultimate resistance, the breaking point, but solely by the location of the yield point, that point at which the material begins to change its shape permanently. A lathe, a machine, a bridge or boiler, once it begins to change its shape permanently, is ruined. It has become the custom in this country to run testing machines at such speed that no one can tell whether the beam is floating at zero and indicating the load that is transmitted to the test piece, and I am ready to prove that in court or anywhere else. Under such conditions it is absolutely impossible to determine the yield point or any other facts. The elastic limit is something we need not talk about, because it is difficult to determine, except by the most sensitive apparatus. The method here described is absolutely inaccurate. I will tell you why. When you determine the one thousandth of an inch of elongation it can only be done by applying a load to the test piece and taking a reading by very delicate apparatus; it must read to the ten-thousandth of an inch in order to get accurately the thousandths of inches. When you take a reading and stop the load and then reload that material, it begins to stretch slightly, but the yield point will thereby be raised.

I wish to prevent such a report going into print. What I am stating are well known facts.

Therefore, I do not want such specifications proposed when there are methods for determining the yield point accurately—by simply running the testing machine at a proper speed. I repeat, that by running a machine as rapidly as stated, no one can know whether the beam is kept floating by the loads applied or by inertia, and I object most strongly to such statements appearing at this late date in a report of this Society.

DISCUSSION.

Mr. Gus. C. Henning.—Proposed specifications treating of the subject named in the report of this Special Committee, and practically identical with it were proposed for discussion at the Boston meeting of this Society, and there and then received a rather thorough discussion which was not controverted.

At the Saratoga meeting the chairman of this Special Committee made a verbal report which again called forth criticism which has not been proven to be incorrect by the Committee in its present report.

On the other hand the chairman makes complaint that he received but slight assistance and scant courtesy from the supposedly interested membership, by reporting as follows: "In the usual way copies of these specifications were sent to various members of the Society, with the usual result—that is, in a few cases, after writing two or three letters, replies were received." It may be necessary to point out at this time and place that the Committee was appointed to develop neither new specifications nor new methods of testing, but merely to evolve from existing knowledge and specifications a new set based on those in use, from which would be eliminated their incompatible differences or incongruities, or clauses which had become unsuitable or useless, and on the other hand to bring all the requirements up to date.

In order to do this a committee should be composed of engineers who are intimately familiar with the subject submitted to them; they should have all specifications before them; be familiar with the design and construction of the products covered by these specifications, and should also be in close contact with the steel works and shops in which the steel is made, the work done and the material tested.

Only such engineers can properly prepare specifications which will be generally acceptable.

In spite of this complaint about lack of co-operation, the committee failed to avail themselves of previous criticism of the specifications which they used as a basis for their work.* The first criticism that I must make is about the laxity and indefiniteness of the language, and the errors and misunderstandings which this necessarily introduces, because if there is anything an

* See pages 642–657, Vol. XXIII., *Transactions*, A. S. M. E.

acceptable specification should do, it is to *specifically specify detail*, to avoid accidental or deliberate misinterpretation.

On the second page of the report which refers to castings, it is stated that 2-inch test-pieces should only be used "when it is inconvenient to use the 8-inch." Now it is a well-known fact that it is never inconvenient to cast one or more 8-inch test-pieces with the casting when only one or several of them are made. It can only be inconvenient to the manufacturer when it does not suit him to do so and in no other case.

These specifications are to be followed not after the work has been done, but are furnished with the call for bids and are supposed to be followed from beginning to the end of the work. Hence there cannot possibly be any excuse for failure to provide 8-inch test-pieces gated from the castings at the time the latter are poured.

On the third page of the report which refers to boiler plate test-pieces, it is prescribed that the "width of specimen along the parallel section shall be 1½ inch whenever possible." This is simply an absurd concession! Every boiler plate is many inches wide, never under 24 inches, and as the specimens are cut out of the crop ends sheared off the full width of plates there never can be any difficulty of obtaining specimens of the prescribed width! *It is always possible to obtain specimens 1½ inches wide in the parallel part*, except when the rolling mill does not want to do so; nowadays tests are invariably made at the rolling mill. Again it is specified that: "When it is inconvenient to use the standard test specimen, it may be made as shown in Fig. 93," which means that a specimen of ½-inch diameter and 2-inch gauge length may be used. How obliging the committee intends to be to the largest mills who roll the largest plates ¾-inch thick and over! No such test-pieces as shown could be cut from thinner plates—just examine the dimensions! On the fifth page of the report a "drop test" is suggested for "small and unimportant castings." Now what is the use of testing this class of castings at all? Moreover there is not one drop-test apparatus in existence at any steel foundry in this country proper for this kind of test. At only a few can very large drops be found, ample to break up very large castings before recharging them in the furnace. A hand-hammer or sledge would seem to me to be more appropriate for the purpose. But what is the use of testing "*unimportant castings*" at all? Do standard specifications ever refer to unimportant material?

On the other hand, the "*percussion test*" is prescribed for "*large castings*"; these "are to be suspended and hammered all over."

Martens * defines what a "percussion test" is supposed to be by those familiar with it—it is a very different test from that referred to in the specifications which is everywhere—the world over—known as the "hammer test."

The *hammer test* again is only useful in case of small castings, and does not have any effect on large ones. How then can it be of any service whatever for the purpose proposed?

Again, how can crushing castings in the large drops prove them to be "suitable for the purposes intended." Such language in specifications is rather ingenuous!

On the fifth page, under "Homogeneity Test," it is stated that "a sample from a broken tensile test specimen shall not show any single seam or cavity more than $\frac{1}{4}$ inch long in either of the three fractures obtained as described below." Now it is well known that every firebox sheet is sheared on all sides, hence if there are any defects in any sheet a competent examination of its edges will always reveal them and should cause its rejection. Even if the plates containing pitting or gas holes in the plates showed only under a magnifying glass this would be ample cause for rejections! What is the use of looking for defects $\frac{1}{4}$ inch long with a magnifying glass, I should like to know. They can be seen by the naked eye at a distance of five feet! Let us examine the method proposed for finding such $\frac{1}{4}$-inch defects and at the same time note carefully the language used which prescribes the use of a "pocket lens" for finding $\frac{1}{4}$-inch defects!

"A portion of the broken tensile specimen is either nicked with a chisel or grooved on a machine, transversely about a sixteenth of an inch ($\frac{1}{16}''$) deep, in three places about two inches ($2''$) apart. *The first groove should be made on one side, two inches* ($2''$) *from the square end of the specimen;* the second, two inches ($2''$) *from it on the opposite side;* and the third, two inches ($2''$) from the last, and on the opposite side from it. The test specimen is then put in a vise, with the first groove about a quarter of an inch ($\frac{1}{4}''$) above the jaws, care being taken to hold it firmly. The projecting end of the test specimen is then broken off by means of a hammer, a number of light blows being used, and the bending being away from the groove. The specimen is broken by the other two grooves in the same way. The

* "Martens's Handbook of Testing Materials," pp. 291–293.

object of this treatment is to open and render visible to the eye any seams due to failure to weld up, or to foreign interposed matter, or cavities due to gas bubbles in the ingot. After rupture, one side of each fracture is examined, a pocket lens being used if necessary, and the length of the seams and cavities is determined."

It will be noted that the material is to be either "nicked with a chisel or grooved on a machine, transversely about $\frac{1}{16}$ of an inch deep," and the effect of this nicking on the steel is not taken into account and is supposed to be inappreciable in either case, even when plates vary in thickness from $\frac{1}{8}$ inch to over 1 inch thickness.* Everyone knows that very different results are produced. But let us look further. It is specified that the grooves are to be made in three places, two inches apart; "the first groove on one side two inches from the square end of the specimen; the second two inches from it, on the opposite side; and the third two inches from the last and on the opposite side from it."

It is at once evident that two nicks will come, when following the instructions, two inches from the square end of test piece and opposite each other, and the third four inches from the end, while it may be clear that this was not at all intended to be the case.

But let us proceed and examine the method further which prescribes that the specimen shall be "broken off by means of a hammer, a number of light blows being used, after it has been put in a vise, with the first groove about a quarter-inch above the jaws, with the additional wise caution, "care must be taken to hold it firmly."

Let us remember that the ends of these test pieces are from $\frac{1}{8}$ to one inch thick and two inches wide, and only three inches long according to Fig. 92, or $\frac{3}{4}$-inch diameter and one inch long according to Fig. 93, when the former shape "is inconvenient."

Now I will challenge anyone to break off a piece of fire-box plate one inch thick by two inches wide, scored $\frac{1}{16}$ inch deep, as prescribed, when clamped firmly in a vise, by means of light blows of a hammer. This is a ridiculous and absurd direction which must be patent to all. This done, it is prescribed that "the specimen *is broken by the other two grooves* in the same way." Has such English ever before been used in Standard

---

* These specifications cover $\frac{1}{8}$-inch plates, as they are referred to on seventh page of the report.

Specifications? The next important point which I am bound to again criticise is the proposed omission of the determination of "yield point," and definition and method of determination of "elastic limit."

It is proposed * "that the determination of the yield point for ordinary grades be omitted."

The use of the words "*ordinary grades*" has undoubtedly been resorted to to ward off criticism.

Let us look at the meaning of the word "*ordinary*." The small boy applies this word to a useless dog or to another boy for whom he has no respect because possessing bad habits and qualities. He has no use for such material. Just the same with boiler plate. What's the use of determining its yield-point, when it's just ordinary, no good, worthless! That's about the idea of the Committee.

But the dictionary defines "ordinary" as follows: "of common or ordinary occurrence, customary, usual." Hence, as these specifications are supposed to apply to all usual, customary or common steel of ordinary occurrence, this word does actually apply to all good boiler steel except that of extraordinary qualities. Hence the committee proposes to drop the determination of the yield point in all cases of testing standard qualities of boiler plate, instead of only of the inferior grades which are unworthy of consideration, as the Committee would have us believe.

Now let us see what yield point is and how it is determined easily and accurately by simplest means. Our honorary member, Professor Unwin, is one gentleman who tells us what the yield point is and how important it is to determine it.† Un-

---

* Second page of the report, paragraph 2.

† "6. *The Yield Point.*—In iron and steel, and in some other rolled or hammered materials, at a stress exceeding more or less the elastic limit, there occurs a large and almost sudden increase of deformation in the ordinary method of testing, and the deformation is permanent, or plastic deformation. For greater stresses the plastic deformation increases, and it amounts, before fracture is reached, to many hundred times the whole elastic deformation. The point at which this almost sudden augmentation of plastic deformation occurs is termed the yield point or breaking down point. It is obvious that a general plastic yielding of a structure would ruin it for practical purposes, hence the yield point seems to fix a limit of stress independent of that determined from considerations of safety against fracture which the working stress should not exceed. The yield point is raised by loading which exceeds the primitive yield point, but it is not usually practicable to raise the yield point of a material artificially

win says, p. 63, "Next to the yield point the most important point to observe is the point where the maximum load is reached."

Professor Martens * is another authority who emphasizes the importance of the determination of yield point and defines clearly the difference between it and elastic limit. Martens says, p. 28, "The yield point as also the proportional limit are well-defined points." Again we may refer to our late member, Prof. J. B. Johnson, who adapted his ideas to those proposed by the French Commission on Methods of Testing Materials, but unfortunately did not interpret the statements published in their language correctly. He translates the French term "la limite d'Elasticité Apparente" as the "apparent elastic limit," and then defines it as a point which is not at all apparent, but can only be determined and recognized by most delicate apparatus and with great care and much labor. The very meaning of the word "apparent" is as follows: "clearly perceived, or perceivable; easily understood, evident." The fact is that according to the definitions of Bauschniger, Martens, Tetmayer, Bach, Unwin, and many others it is readily observable by any careful inspector or engineer in a mill, and does not require a laboratory equipment with most highly trained assistants.

Why this method proposed by Johnson for determining the "elastic limit" (when his French models distinctly state that they coined the expression to indicate the "yield point" for which they had no word) and re-stated by Kent in Trans. Amer. Inst. Mining Engrs., 1903, should now be adopted by this committee for determining a doubtful point very difficult to ascertain, is difficult to understand. Let us examine what this method proposed by Johnson and adopted by Kent really means and leads to. Martens says on p. 30:

"This is the proper place to call attention to a *very important misconception* which is produced by the uncertainty of accurate definition of the idea of elastic limit, and the existing careless distinction between proportional and elastic limits and of yield point."

If you will read the references to Unwin and Martens, given heretofore, relating to yield point, you will find that adding loads

before using it in a structure, and consequently the primitive yield point, due to the mechanical operations of manufacture, fixes with respect to deformation the dangerous limit of stress." "The Testing of Materials of Construction," W. C. Unwin, pages 7, 62, 93, 99, 250, 305.

* "Hand-book on Testing Materials," Martens, pages 28, 30, 44, 261, etc.

successively and intermittently always raises it when testing materials like boiler plate. Another point is this, that the yield point shows itself very suddenly in many cases within an increment of load of 250 pounds per square inch.

The plates which these Specifications refer to vary from $\frac{1}{8}$ inch to over one inch in thickness as the allowable variations in weight of all of these thicknesses are given in extensive tables. The proposed increments of load for consecutive measurements of extension are 1,000 pounds per square inch. Now the sections of the proposed test-pieces $1\frac{1}{2} \times \frac{1}{8}$ inch to $1\frac{1}{2} \times 1$ inch will vary from 0.18 to 1.50 square inch.

The necessary load on the testing machines to produce 1,000 pounds per square inch on 0.18 square inch section will be 180 pounds. It is a practical impossibility, therefore, to make such a test on thin plates with any ordinary machine, and it is impossible to make it accurately on thicker test-pieces with any machine running at the speeds customary at the present day in all mills. The mere starting and stopping of such a machine will add a thousand pounds more or less to an indefinite degree to my own knowledge, by the great inertia of machine at such speeds and the inertness of all operators under such conditions.

Moreover, the time required for such determination of what is erroneously called "elastic limit," but should have been called "yield point," is so great and troublesome that no mill, where the testing is to be done according to these specifications, would or could put up with it. It would simply paralyze all the mill-testing laboratories in the country. Moreover, the yield point makes itself as quickly apparent on a large as well as on a small test piece, because when this point is reached the material seems to break down instantaneously.

The fundamental rule for making all tests of materials is to run the machine so as to add equal increments of load continuously in equal intervals of time. The above method is absolutely opposed to this fundamental rule necessary to obtain uniform and comparable results.

There is one simple and accurate way of determining the yield point, which should be done in every case—as the yield point is the most important point to be fixed according to all authorities, for the proper design of all machines and structures subject to varying loads.

This simple and accurate method is to use a pair of finely

pointed dividers, with a reading glass mounted on one leg to show accurately what is happening under one of the points.

The dividers are set to any distance about eight inches. When the dividers have then been set on the test piece with the point of one leg in a punch mark, a fine scribe-line is made on the test piece with the other. As the loading then proceeds, stretching at a uniform rate will become apparent as the scribed line moves away from the point of the dividers. Instantly the yield-point is reached the rate of extension increases with remarkable speed in a striking manner and the operator then takes a reading of load without any disturbance or interruption in the operation of the test.

This method is quite within the capacity of all those usually employed in making routine tests in shops and mills, and gives accurate results wherever desired.

There is another suggestion I should like to make to the committee as it has had such difficulty in obtaining assistance in preparing these proposed standard specifications.

I refer to Figs. 92 and 93, in which account has been taken of metric dimensions * after this Society last spring decided to have nothing to do with the metric system, as it was no good at the present time, in fact being hardly in general use anywhere.

These figures show metric dimension to the hundredths of a millimetre. A hundredth of a millimetre is not quite .0004 inch, and our mechanics and those using the metric standards might become perplexed and get these dimensions incorrect to one of two hundredths of a millimetre. Had not these fractions better be left off?

Moreover, if the approximate metric measures were substituted, the principal dimensions of the proposed test-pieces would be almost identical with those used in continental Europe. But I have little hopes that our committee will drop a decimal of a millimetre in the radius of a fillet or in the total lengths and dimensions as shown. Moreover, there is one positive error in Fig. 92. There should not be eight divisions of the 8-inch gauge length, but twenty of 0.40 inch each, which is practically the same as the centimetre divisions which the Europeans have adopted. The one-inch divisions are much too large for measuring the proportional elongation of the test-piece, and those 0.40 inch are just about right, and results measured thereon can be

* Editor :—These metric dimensions have been omitted in revision.

readily compared with those obtained in Europe or published in the most valuable foreign reports, and without introducing any measurable errors.

It is not possible to prepare one set of Standard Specifications to cover boiler plate, rivet steel, steel castings and forgings.

This is demonstrated by the fact that in the report before us specifications for tank or sheet steel have become mixed up with those for boiler plate, as will be seen by comparing the tables of allowable variations in weight of plates. One table refers to boiler plate from $\frac{1}{4}$ inch thickness and over, while the other refers to all sheets $\frac{1}{4}$ inch in thickness and under. It is a well-known fact that such thin plates are never used in boiler construction, hence any reference to them in this paper is quite out of place. Much more might be criticised in the report as to the indefiniteness as to how many tests shall be made, what facilities for making them shall be provided, and who shall provide them.

In these Specifications everything referring to these matters is left entirely to the good will of the steel maker, and no authority is given to the purchaser of the material, who is the first party to be satisfied.

It is to be hoped that this review of the report will give the members of the committee cause for reflection, and enable them to produce a report more nearly up to the present time in the field of testing materials and to prepare practical and useful Standard Specifications, which will serve the objects of the engineer as well as those of the steel maker. The apparent result of producing those before us has been a rather sad one.

Mr. W. W. Dingee.—

RACINE, WIS., U. S. A., *November* 14, 1903.

Prof. F. R. Hutton:

Dear Sir: In response to circular No. 979, I would say that for the past seven years the J. I. Case Threshing Machine Company of Racine, Wis., have furnished their Purchasing Agent with specifications for all the various material used in their business. They have a well equipped Chemical and Physical laboratory where these specifications are prepared and where samples of all material received are tested to see that they come up to requirements. These specifications embody the result of our long experience in the requirements of material for our special purposes and as a result we now have control of many subtle influences that formerly were not understood and the tendency of which was to make the life of a manufacturer a burden.

Enclosed are samples of these papers touching the subject matter under consideration and which may be made a part of the discussion.

Yours truly,

(Enclosure.) W. W. DINGEE.

No. 307.

SPECIFICATIONS FOR BOILER RIVETS.

The rivets purchased under this specification are to be made of the best grade of Open-Hearth steel, and should be formed in solid steel dies, *i.e.*, closed dies.

The rivets must be exact in size and fit the metallic gauges furnished by the J. I. Case T. M. Co.

In the event of no gauge being furnished it is understood that the shape and dimensions are to conform to the blue prints furnished by the company, or to a regular stock sample.

The body of the rivet should be perfectly round, exact in size, and free from oxide or scale.

The head should be in accordance with the metallic gauge, blue print, or sample.

The rivets must be free from all injurious defects, be finished in a workmanlike manner, and fulfill the following requirements:

Physical Test.

Cold Bending Nick Test.—Nicked with a cold chisel on one side of the shank or body, the rivet must bend double (away from the nick), flat upon itself (*i.e.*, 180 degrees) without breaking.

NOTE —A hand cold chisel must be used in nicking the test pieces, and a hammer of not over three pounds in weight used for striking the chisel. No rivet will stand the bending test if nicked deeper than one-fourth the diameter of the rivet.

Quench Test.—Heated to a cherry-red, quenched in water at a temperature of 80 degrees Fahr., and then nicked on the side with a cold chisel, the rivet must bend double (away from the nick), flat upon itself without breaking.

Disk Test (Cold).—A rivet held in an upright manner or vertical position, under the trip or drop hammer, and flattened by repeated blows to a circular disk must show no signs of "cold shortness," *i.e.*, must not crack, split or crumble.

Flat Test (Hot).—A rivet heated to a bright cherry-red and placed in a flat or horizontal position under the hammer, must show no signs of "red shortness" (*i.e.*, must not crumble), when flattened to a thickness of one-fifth the original diameter of the body.

Chemical Composition.

Rivets which will pass all of the required physical tests should be within the following limits in regard to composition:

Phosphorus	should	not	exceed	0.03	per	cent.
Sulphur	"	"	"	.025	"	"
Silicon	"	"	"	.02	"	"
Manganese	"	"	"	.50	"	"
Carbon	"	"	"	.15	"	"

Remarks.—Material will not be accepted which fails to meet the requirements in regard to size, uniformity and gauge dimensions.

Failure of more than five per cent. of the rivets to pass the physical tests will be cause for rejectment.

(A certain number of rivets will be taken from each keg or package for the physical test, and such test decides the acceptance or rejectment of that package in particular, but if more than half of the individual tests fail, the whole lot may be rejected.)

Material will also be rejected which shows on analysis:

Phosphorus	over	0.04	per	cent.
Sulphur	"	.035	"	"

Any alteration in this specification, either in the physical requirements or the chemical composition, must be stated in writing at the time of contract and signed by the party furnishing the material, also by the Purchasing Agent.

No. 302.

SPECIFICATIONS FOR STEEL CASTINGS.

Steel for castings may be made by either the Open Hearth, Crucible, or Tropenas Bessemer Process; preference being given the open-hearth product.

The castings must be true to pattern, free from external blemishes, blow holes, shrinkage cracks, cold shuts and other injurious defects.

No porosity shall be allowed in portions where the resistance and value of the casting, for the purpose intended, will be seriously affected thereby.

The castings are to be annealed or un-annealed, as specified by the purchasing agent under the head of remarks.

If annealed, the castings must receive a proper heat treatment and be given a sufficient time in which to cool, so that all internal strains are relieved and the metal assumes its proper degree of ductility.

Large castings suspended and hammered all over, should not after this treatment show any defect or weakness.

The castings must be properly cleaned and free from sand, scale, etc.

Steel castings will be divided into three classes: Soft, Medium and Hard.

The physical qualities and chemical composition of the different grades or classes must conform to the following requirements.

Soft Cast Steel.

Physical Test.

Test pieces for the physical test may be cut cold from a coupon, molded and cast on some portion of one or more castings from each melt, or from a sink head, riser or sprue, on such castings.

The test piece should be fashioned so as to give a gauged length of two inches and about one-half inch diameter.

A bending test made on a specimen one inch wide and about one-half inch thick, must bend cold, around a diameter of one inch, through an angle of 120 degrees without fracture on the outside of the bent portion.

Tensile Strength must not be less than 58,000 pounds per square inch.

Elastic Limit must not be less than 27,000 pounds per square inch.

Elongation measured in 2 inches, must not be less than 22. per cent., and the reduction in area should not be less than 30. per cent.

Chemical Composition.

Carbon	may range	from	0.10	to	0.22	per	cent.
Manganese	"	"	.20	"	.75	"	"
Phosphorus	must	not	exceed		.08	"	"
Sulphur	"	"	"		.06	"	"
Silicon	should	not	exceed		.40	"	"

Medium Cast Steel.

Physical Test.

Tensile Strength must not be less than 65,000 pounds per square inch.

Elastic Limit must not be less than 31,000 pounds per square inch.

Elongation measured in 2 inches, must not be less than 18. per cent., and the reduction in area should not be less than 25. per cent.

Chemical Composition.

Carbon	may range from			0.22 to	0.35	per	cent.
Manganese	"	"	"	.25 "	.80	"	"
Phosphorus	must	not	exceed		.07	"	"
Sulphur	"	"	"		.05	"	"
Silicon	should	not	exceed		.35	"	"

Hard Cast Steel.

Physical Test.

Tensile Strength must not be less than 80,000 pounds per square inch.

Elastic Limit must not be less than 39,000 pounds per square inch.

Elongation measured in 2 inches, must not be less than 12. per cent., and the reduction in area should not be less than 20. per cent.

Chemical Composition.

Carbon	may range from			0.35 to	0.50	per	cent.
Manganese	"	"	"	.30 "	.85	"	"
Phosphorus	must	not	exceed		.06	"	"
Sulphur	"	"	"		.04	"	"
Silicon	should	not	exceed		.30	"	"

When no special grade or class is specified by the purchasing agent, the material may be anywhere within the limits of a soft and hard steel.

NOTE.—It will be noticed that the specified chemical composition is not at all rigid except in the case of phosphorus and sulphur.

Experience has shown that phosphorus in unworked steel, if very high, produces extreme brittleness by making the material "cold short."

High sulphur produces "red shortness," increases shrinkage, makes the casting hard, and indirectly causes blow holes.

Manganese stiffens the steel, raises the elastic limit, partially neutralizes the effect of sulphur, eliminates the occluded gases, and in a measure prevents blow holes.

Silicon in the initial charge raises the heat and imparts fluidity; in the final product it has a tendency to impart rigidity and hardness.

Aluminium, either pure, or in the form of "Ferro-aluminium" is generally added to the melted steel for the purpose of quieting the bath, raising the heat, purifying the metal, and imparting greater fluidity; an excessive amount of aluminium added to a steel low in manganese, will often produce a segregated porosity in the material.

Blow holes in steel castings are the most common cause of trouble, and the defect is seldom discovered until the casting breaks.

Porosity or spongy places in castings are also a source of trouble; this may in a measure be avoided by the use of fillets in sharp corners and angles, and by changing the form of the pattern so as to relieve the draw.

Under the head of physical test we have allowed an elastic limit of less than half the ultimate tensile strength, usually however, it will be higher, and in the high carbons, somewhat closer to the breaking point.

Carbon is not rigidly specified, as noticed in the adoption of the word "should," which implies that the maker be allowed a reasonable variation in order to get the desired strength.

We reserve the right to use the drop test or any other reasonable test to determine the quality of the material and its freedom from blow holes or other flaws.

Instruction in regard to annealing, and such changes as the purchasing agent may decide on in respect to physical requirements, chemical composition, etc., must be stated in writing and signed by the party in question.

No. 324.

SPECIFICATIONS FOR STEEL RIVET RODS.

Two grades of steel are considered in this specification and are to be designated as "Extra Quality" (Class A), and "Ordinary Quality" (Class B).

Class A is to be a mild steel of superior quality, made by the Open-Hearth process.

Class B is to be a good grade of soft steel, made by the Open-Hearth or Bessemer process, as ordered and agreed upon at the time of contract.

Both grades of material must be free from injurious defects, be free from excess of scale, finished in a workmanlike manner, and conform to the requirements of the physical test and the chemical composition.

"Extra Quality (Class A)."

Chemical Composition.

Phosphorus	must not exceed	0.03	per cent
Sulphur	" " "	.025	" "
Manganese	should not exceed	.50	" "
Carbon	" " "	.15	" "
Silicon	" " "	.02	" "

Material will not be accepted which shows on analysis:

Phosphorus	above	0.04	per cent.
Sulphur	"	.035	" "

Physical Tests.

The *Tensile Strength* should not be less than 55,000 pounds or more than 62,000 pounds per square inch.

The *Elastic Limit* must not be less than one-half the ultimate strength.

The *Elongation*, measured in 8 inches, should not be less than 28.00 per cent.

Material will not be accepted which shows a tensile strength of less than 45,000 pounds or more than 65,000 pounds per square inch.

"Ordinary Quality (Class B)."

Chemical Composition.

Phosphorus	must not exceed	0.09	per cent
Sulphur	" " "	.05	" "
Manganese	should not exceed	.50	" "
Carbon	" " "	.15	" "
Silicon	" " "	.05	" "

Material will not be accepted which shows on analysis:

Phosphorus	above	0.10	per cent.
Sulphur	"	.06	" "

Physical Tests.

The *Tensile Strength* should not be less than 52,000 pounds or more than 62,000 pounds per square inch.

The *Elastic Limit* must not be less than one-half the ultimate strength.

The *Elongation*, measured in 8 inches, should not be less than 26.00 per cent.

Material will not be accepted which shows a tensile strength of less than 45,000 pounds or more than 70,000 pounds per square inch.

Cold Bending Test.—Class A and B material must bend flat upon itself (180 degrees) without showing signs of fracture on the outside portion of the bend.

Quench Test.—Class A or B must show no signs of fracture when bent flat upon itself, after heating to a cherry red and quenching in water.

Hammer Test.—Class **A**, material heated to a bright cherry red and drawn out under the hammer to a thickness, at the point, to one-fifth its original diameter must not split, crack or crumble.

Class B, treated in the same manner must not crack or crumble when hammered to a thickness equal to one-third the original diameter of the rod.

Any alteration in this specification in regard to composition or physical requirements pertaining to either class of material must be stated in writing at the time of contract and signed by the Purchasing Agent and the party furnishing the material.

No. 287.

SPECIFICATIONS FOR BOILER PLATE, FIRE BOX AND FLANGE STEEL.

Boiler Shell Plates, Front Tube Plates and Butt Strips.

The material purchased under this specification is understood to be a superior grade of "Open-Hearth" steel.

It must be homogeneous in structure, free from blister, cracks and other injurious flaws or defects.

It must also fulfil all the requirements of the following physical test:

Physical Test.

Tensile Strength.—The ultimate breaking strain must not be less than 52,000 pounds or more than 65,000 pounds per square inch.

Elastic Limit.—The elastic limit must not be less than one-half of the ultimate tensile strength.

Elongation.—The elongation measured in eight (8) inches, must not be less than the following per cent., according to thickness:

20 per cent. for plate $\frac{3}{8}$ in. and under.
22 " " " " $\frac{3}{8}$ to $\frac{3}{4}$ in. thick.
25 " " " " $\frac{3}{4}$ in. and over.

Cold Bending Test.—The specimen must bend flat upon itself (180 degrees), without showing signs of fracture at any part of the bend.

Quench Test.—A piece of the steel plate heated to a bright cherry-red color, then quenched in water at a temperature of

82 degrees Fahr. must bend around a curve equal to one and one-half ($1\frac{1}{2}$) times the thickness of the plate without showing signs of fracture on the outside portion of the bend.

Chemical Composition.

In order to meet all the requirements of the physical test and be within the limits of a superior grade of Open-Hearth steel, the chemical constituents must conform to the following specified percentage:

Carbon, not under 0.12 or above	0.20	per cent.	
Phosphorus, not above	.04	"	"
Sulphur, " "	.04	"	"
Manganese, " "	.50	"	"
Silicon, " "	.05	"	"

Rejectment.—Material will be rejected which shows:

(1) A tensile strength of less than 50,000 pounds per square inch or more than 65,000 pounds unless the elongation is 28 per cent. or more.

(2) An elongation less than the quotient of 1,400,000, divided by the tensile strength per square inch.

(3) Failure to pass the cold bending or the quench test.

(4) Phosphorus above 0.05 per cent. Sulphur above .05 per cent.

Fire Box and Back Tube Plate.

The material desired under this heading is understood to be the best grade of "Open-Hearth" steel that it is possible to make by modern methods.

Metal of the following composition is desired.

Carbon,		0.18	per cent.	
Phosphorus,	not above	.03	"	"
Sulphur,	" "	.02	"	"
Manganese,	" "	.40	"	"
Silicon,	" "	.02	"	"

Material will be rejected which shows on analysis:

Carbon,	below 0.14 or over	0.25	per cent.	
Phosphorus,	above	.035	"	"
Sulphur,	"	.045	"	"
Manganese,	"	.50	"	"
Silicon,	"	.05	"	"

Physical Test.

This steel should pass all the specified tests, viz:

Tensile Strength.—The ultimate breaking strain should not be less than 52,000 pounds per square inch, nor more than 65,000 pounds.

Plates with a tensile strength of more than 65,000 pounds will not be rejected providing the elongation is 30 per cent. or over.

Elastic Limit.—Must not be less than one-half of the tensile strength.

Usually the elastic limit is about 35,000 pounds where the tensile strength averages 60,000 pounds per square inch.

Elongation.—Measured in 8 inches, should not be less than 28 per cent; and in all cases it should not be less than the quotient obtained by dividing 1,450,000 by the tensile strength per square inch.

This material must also stand the cold bending test and the quench test.

It must also be free from seams, cracks, cavities due to gas bubbles in the ingot, pipe laminations or other defects.

A micro-photograph of the material should show a homogeneous structure.

The plate should be of even thickness, straight, smooth rolled and finished in a workmanlike manner.

Mr. Walter Flint.—

New York City, *November* 18, 1903.

American Society of Mechanical Engineers:

Gentlemen: I have the report of the committee which was appointed to make out specifications for boiler plate, rivet steel, etc., and I have looked the matter over quite thoroughly. It seems to me that Mr. Henning's point is well taken. Surely every one must admit that when a member of a bridge or boiler or any other structure is loaded beyond the "yield point," it will never again perform its duty in that structure, and the other members are then called upon to perform a duty for which they were not designed. I will, therefore, say that the determination of the "yield point" certainly ought to be retained, and every care taken to determine it with accuracy.

Yours very truly,

Walter Flint.

L. S. Randolph.—This discussion seems to be open to criticism for the following reasons:

First. The grouping of the specifications for steel castings and boiler plate is a very decided mistake. This material is not used by the same departments of the shop, or are they made in the same shop, and it is a useless complication to have one specifi-

cation covering two classes of material so widely different as boiler plate and steel castings. It is doubtful whether there is any reason for the riveted steel going on the same sheet with boiler plate.

Second. The minimum requirements should be lower with possibly a different nomenclatum carrying it very soft and giving 55,000 pounds ultimate tensile strength plus or minus 5,000 pounds. This allows material to come under this head which is needed for certain classes of fire-box work. Of course, the corresponding change in the elongation being made.

Third. The specifications are a little ambiguous as to the size of test specimens, the indication being that 2½ inch test specimens can be used on boiler plate, although this may be intended to castings and forgings only. One of the most important considerations in testing materials is to have the specimens of uniform size. If there is much work where the large specimens cannot be obtained, a small specimen should be adopted as standard and should be held to.

Fourth. It should be stated that the facilities for making tests, etc., should be furnished by the manufacturer free of cost. This is what is being done at the present time.

Fifth. I agree with Mr. Henning in the dropping of the yield point. It would be much more sensible to drop the ultimate tensile strength and leave the yield point in even if we had to slow down our machines somewhat. There is little doubt in the mind of the writer that much of the inspection to-day is simply perfunctory. The specifications should be drawn up primarily so as to insure the furnishing of first-class material, and should then be eased off so as to get everything that is absolutely necessary in the quality of material desired at a minimum price and at the minimum expenditure of trouble by the manufacturer. The manufacturer, however, should be made to come to the specifications, and not the specifications to the manufacturer.

Sixth. The drop test and percussion test are, in the opinion of the writer, hardly necessary for the boiler steel, and the specifications are so ambiguously drawn in regard to the two tests that it is impossible to say whether they are meant for castings only or boiler steel also. It should be said that it is very difficult to have specifications so widely different in uses and characteristics under the same heading and not have such ambiguities occur.

A. Bement.—A most surprising feature of these specifications is that the determination of the most important characteristic is excluded, that of the yield point. I cannot believe that the members of this Society will allow the perpetuation of this grave error in the final report of the committee. It is specified that this omission be made for "ordinary grades," which means, in fact, all commercial grades, and is almost equivalent to recommending the abandonment of this determination altogether.

I would suggest that requirements for material used for bolts and studs be included in these specifications, especially those employed in engines and similar machinery. With rivets, if unsuitable material is employed, there is liability of their breaking before delivery to the customer. But with bolts and studs, bad material can be used without liability of failure until delivery of the machine is made to the customer; then failure is quite liable to be attributed to some other cause.

Prof. G. Lanza.—I should like to make a few remarks, simply to touch upon two points. Professor Spangler, in proposing to drop the yield point, claimed that in steel varying from 45,000 to 65,000 pounds tensile strength the other requirements would secure, in every case, a yield point of at least half the tensile strength. In making that remark he admits its importance, and it seems to me that the yield point requirement ought not to be dropped, at any rate, unless that position is indisputably proved, and I do not believe that it is proved yet beyond the possibility of doubt.

Another point is that, in considering axles and moving parts of machinery, which are subjected to alternate stresses, we need to look after our specifications with reference to the power of material to bear repeated stresses. Investigations upon repeated stress show that the yield point plays some part which is not very well known yet, and it seems to me important that we should retain a requirement for it until these relations can be determined.

The only other remark I desire to make is, that when too high a speed is adopted in the testing machine, it is not only the yield point which is concealed, but various other things which are of importance—matters which concern the tensile strength also.

Prof. R. C. Carpenter.—I would like to make a remark on this. It seems to me that the yield point is of importance, and of suf-

ficient importance to be retained in these specifications. One of the important things which I think may result from retaining the requirement for determining the yield point is that it will lead to more care in the testing of materials. At the present time the material tests made by manufacturers are made too rapidly to be accurate, as I have found out by experience. Indeed, they are in many cases very far from accurate. This additional requirement must require more care, and will give us much more reliable results, from commercial tests.

H. W. Spangler.—As the Chairman of the committee has received but one communication, that of Mr. Bement, relating to the matter of these specifications, which discussion is herewith presented, there has been no meeting of the committee since the last meeting of the Society. The following comments are his own, and not those of the committee.

It is to be remembered that these specifications cover steel between 45,000 and 85,000 pounds tensile strength having definite chemical and physical properties.

There has been but one objection raised to these specifications, and that as I understand it is to the dropping of the yield point (or elastic limit) for materials covered by these specifications. With the other requirements of the specification fulfilled, a high elastic limit will be obtained. I would be glad to be referred to any data showing that if all the requirements of this specification are met, the yield point is not as high or higher than half the ultimate strength. A careful examination of the reports of the Watertown Arsenal for 1901 and 1902 shows that there are 133 tests of steel, mostly castings and forgings (the ultimate strength of which would bring them within the limits of these specifications, and which materials might be tendered under these specifications) having the elastic limit less than one-half the ultimate strength. Of the samples one hundred and nineteen would not fulfil the other physical requirements. As the chemical composition of these samples is not stated, it is impossible to tell from the printed reports whether the remainder would fulfil the specifications or not, but 89.5 per cent. would be rejected on their physical tests alone exclusive of the elastic limit and the chemical composition.

The objection made by Mr. Henning that the suggested method of determining the elastic limit (yield point) is inaccurate because " when you take a reading and stop the load and

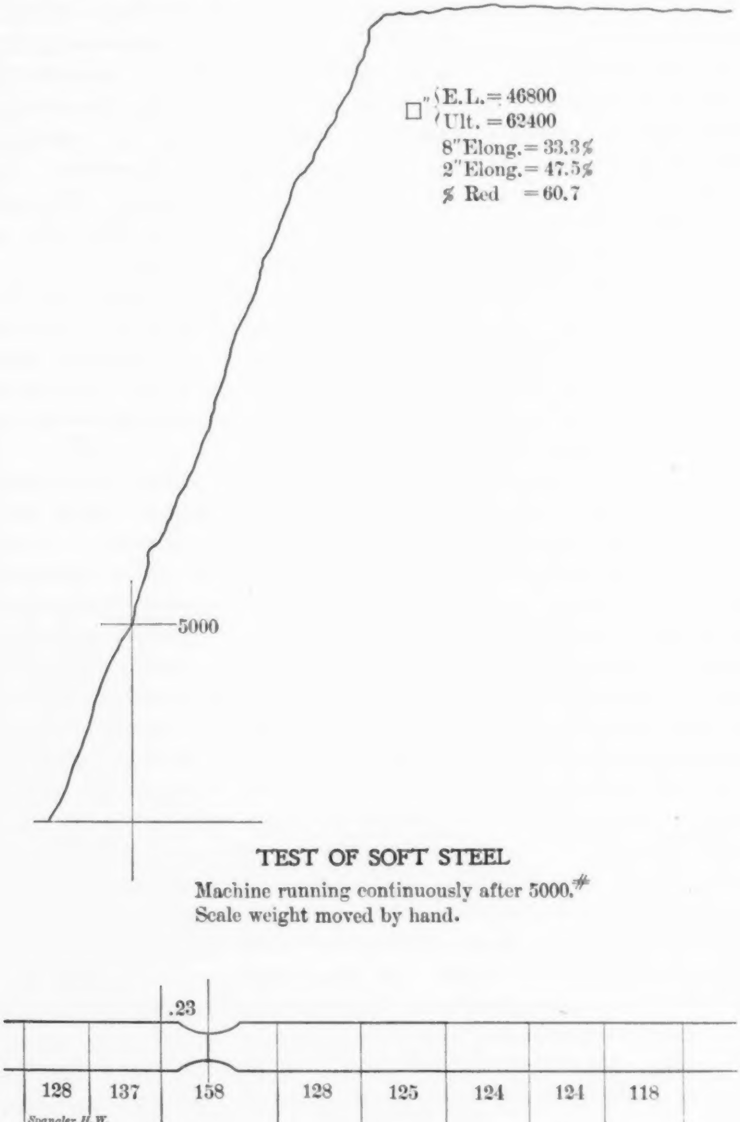


FIG. 94.

then reload the material, it begins to stretch slightly, but the yield point will thereby be raised " does not, I believe, apply to materials that are not loaded above the proportional limit, and

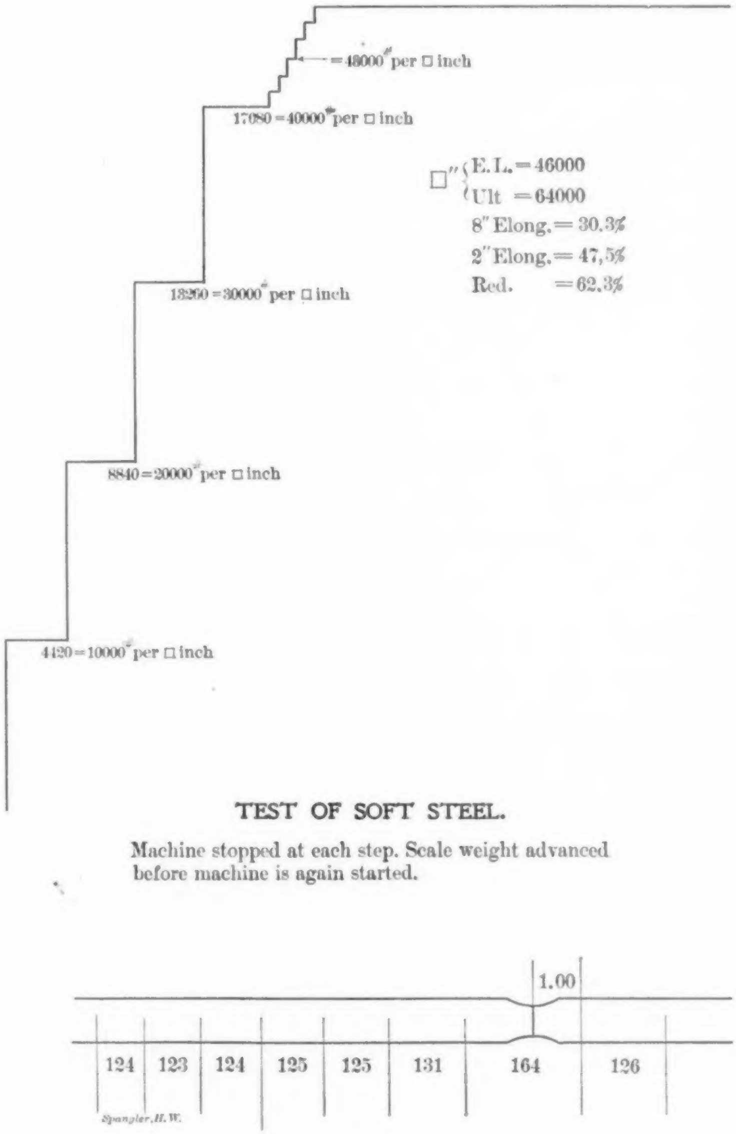


Fig. 95.

practically I do not believe that the value of the quantity to be determined here is affected by the stopping and starting of the machine necessary for the reading of a micrometer extenso-

meter. A series of tests were made by me on a bar of commercial soft steel to test this point. An extensometer having a magnification of about 150 times (exactly 147 times) was used,

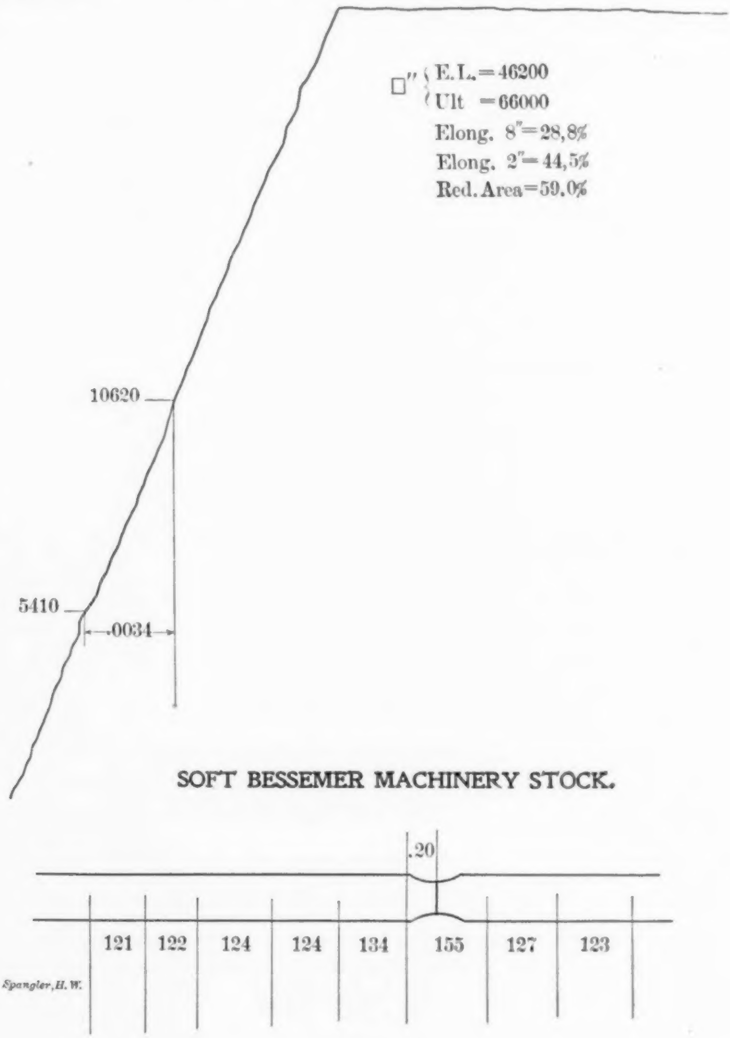
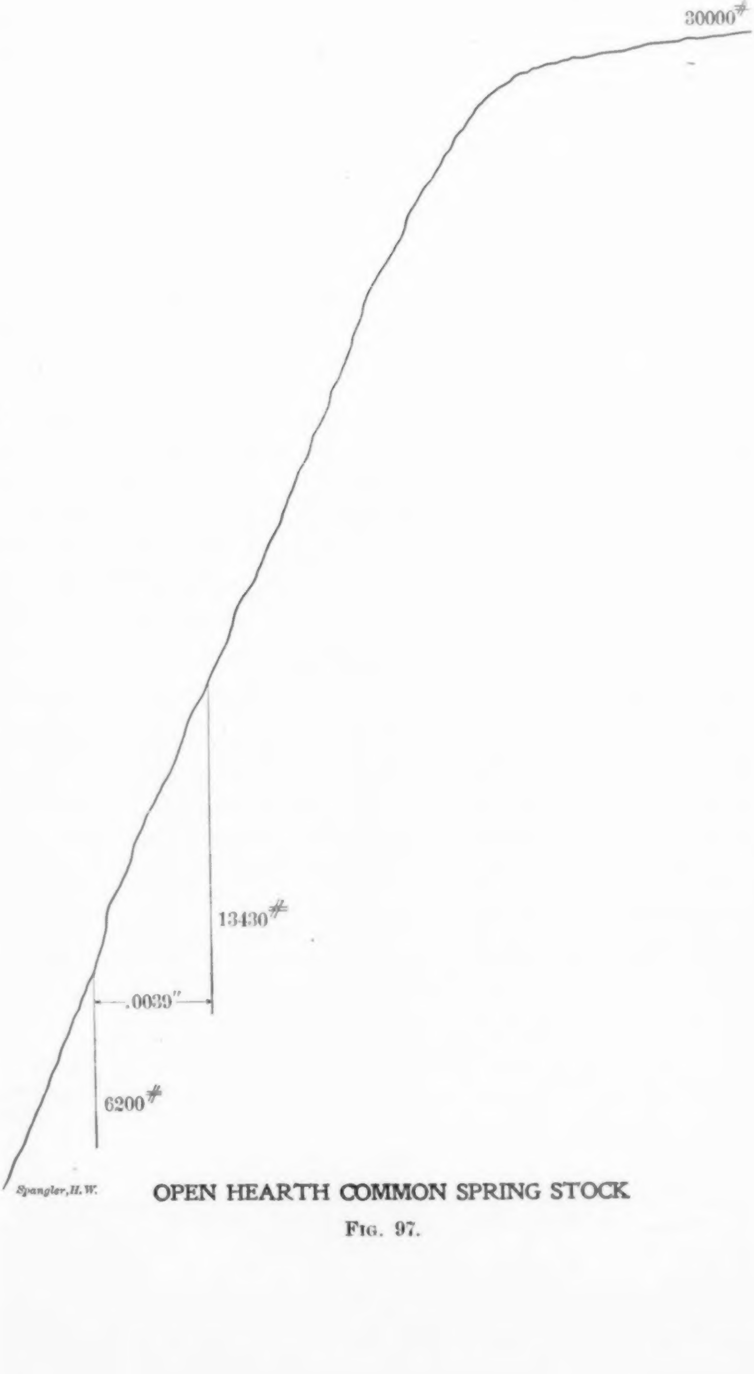


FIG. 96.

and two of the diagrams obtained, together with the results of the eight tests made, are here recorded.

In Fig. 94 herewith the scales are horizontally 147" = 1"



OPEN HEARTH COMMON SPRING STOCK

FIG. 97.

stretch, and vertically 1″ = 4,587 pounds. The irregularity is due to the fact that the weight on the scale beam was moved by hand. In Fig. 95 the scale beam was first set at 4,420 pounds, and the load applied as rapidly as in Fig. 94. When the scale beam rose the machine was stopped and the micrometers were read. The scale beam was then set at the next load and the machine started, each jog in the diagram showing one stoppage of the machine.

While it is true that a few experiments may mean but little, I know of no other printed data covering the same ground and submit these results which show that in this case at least the stopping of the machine a number of times and allowing it to remain at rest long enough to read the micrometers did not perceptibly raise the yield point.

I attach also two test sheets, one from soft Bessemer machinery stock, Fig. 96, and one from open hearth common spring stock, Fig. 97, showing that before and after the machine had been stopped there has been in these two cases no change in the general direction of the curve. In each of the two cases the machine was stopped twice to determine the scale of the diagrams.

Commercial Soft Steel.

Method of applying load.	Ultimate.	Yield point.	Elongation in 8 in., p. c.	Elongation in 2 in., p. c.	Reduction p. c.
No. 6—Step	64,000	46,000	30.3	47.5	62.3
No. 5—Continuous	62,400	46,800	30.3	47.5	60.7
No. 3—Step	61,300	47,000	29.1	49.0	62.3
No. 2—Continuous	61,500	45,300	31.5	50.0	62.3
X—Step	64,100	45,000	Outside marks.		
Y—Continuous	63,300	47,500	28.9	42.0	64.0
Z—Step	62,700	47,000	30.1	46.5	60.7
A—Continuous	64,000	47,300	28.4	47.0	62.3
Average step	63,025	46,250	29.8	47.6	61.8
Average continuous	62,800	46,725	29.8	46.6	62.3

No. 1027.*

ORDNANCE FOR THE LAND SERVICE.

MAJOR R. BIRNIE, ORDNANCE DEPARTMENT, UNITED STATES ARMY.

1. The development of the modern land and naval ordnance in the United States is of comparative recent date, and is a subject for interesting study from a political as well as an industrial and mechanical standpoint. It is with the latter, however, that I have to deal, and particularly the mechanical features, although such is their diversity that an outline only can be attempted. My subject, generally, will be confined to a brief account of the growth of the present land armament, the uses to which the guns and their accessories are applied, and to some discussion of the more important mechanical principles involved in their construction in connection with experiments.

2. There are two general divisions of artillery for land service; first, the mobile artillery designed for field and siege service, and used also in the land defense of sea-coast forts, or generally for offensive operations, in which the weight of the material combined with the specific service in view control the caliber and design; second, the artillery on fixed mounts as used generally in sea-coast fortifications, and also in permanent inland defenses for defensive purposes, in which the caliber and design are not restricted by weight and may be varied to subserve only the needs of defense.

3. The different types of artillery are classified as guns, howitzers and mortars. Guns are used for direct fire with a superior angle of elevation, usually limited to about 15 degrees. They fire a single weight of powder charge with the object of attaining a maximum of power. Howitzers are pieces of medium length and velocity, used for curved fire up to about 45 degrees elevation. They apply generally to mobile armament for field and siege service, and by reason of their relatively

* Presented at a Special Monthly Meeting in New York (February, 1904) of the American Society of Mechanical Engineers, and forming part of Volume XXV, of the *Transactions*.

light weight as compared with guns can be made of larger caliber and use a heavier projectile than a gun of equal manœuvring capacity. This advantage and their supplemental use to guns is further obtained by ability to deliver the curved fire at relatively short range, for which reduced charges may be used. Mortars are short pieces fired with medium or low velocity and used for high angle fire with elevations up to about 70 degrees. They require a fixed platform, and their utility is limited to siege and sea-coast emplacements. Their firing angle enables complete protection to be given them in their emplacements from a hostile gunfire. The projectile carries a large bursting charge, and the angle of fall enables it to reach the most vulnerable parts of targets generally. In the case of a battleship this target embraces not only the relatively thin-armored deck, reached by direct impact, but also the unprotected hull, reached by under-water explosions. To cover the field within the range of a mortar different powder charges and elevations are employed, giving successive zones of fire.

4. The following tables, which include the various calibers of standard types in our service, enable a more comprehensive view to be taken of the diversity of types included in a complete equipment and how they are employed. All of these are rifled, breech-loading pieces, made of steel and use smokeless powder, except the model 1886 12-inch mortar, which is composed of a cast-iron body reinforced with steel hoops. The tables omit the small arm rifle caliber Gatling or automatic machine guns that are mounted on tripod or wheel mounts and used both in the field and in fortifications for combating men and horses.

5. All of the carriages are fitted with hydraulic recoil cylinders and spring return, except the 3.6-inch mortar and the 3.6-inch gun field carriage which is of older pattern, dating from 1891. The 3-inch model 1902, with long recoil of gun on carriage, is of the most modern type. Modifications of the existing mobile artillery now in progress or proposed are as follows:

6. Introduce a light field gun to operate with cavalry, having a caliber 2.38 inches, weight of projectile 7.5 pounds, and muzzle velocity 1,700 f. s., and a 3.8-inch field howitzer of the same mobility as a 3-inch field gun, to fire a 30-pound projectile.

7. Eliminate the 3.6-inch field gun, 5-inch field howitzer, 5-inch siege gun, 7-inch siege howitzer, and substitute therefor 3.8-inch rifles firing a projectile weighing 30 pounds, and 4.7-inch

TABLE I.

FIELD AND SIEGE ARMAMENT.

Piece.	Length of bore.	Powder charge.		Projectile.		Muzzle velocity.	Approximate effective range.	Character of Mount.	Number of horses.	Limiting weight per horse.	Employment.
				Kind.	Weight.						
	Calibres.	*Lbs.*	*Ozs.*		*Lbs.*	*F. s.*	*Yds.*			*Lbs.*	
2.953-inch mountain gun	10.7	....	8	Shell and shrapnel.	12.5	920	4,000	Wheel'd carriage, dismountable	....	......	Material and 100 rounds of ammunition require 13 pack mules per gun.
3-inch field gun. Model 1902..	28	....	26	Steel shell and shrapnel	15	1,700	6,500	Field carriage, long recoil.	6	650	Corps Artillery.
3.6-inch field gun.............	23	....	28	Shell and shrapnel.	20	1,550	5,500	Field carriage.	6	750	Corps Artillery.
3.6-inch mortar...............	5.25	..	6	Shell and shrapnel.	20	690	3,365	Platform carriage.	....	......	Siege operations.
5-inch field howitzer..........	12	....	25	Shell and shrapnel.	55	1,000	6,000	Wheeled carriage.	6	800	Corps Artillery.
5-inch siege gun..............	27	5	6	Shell and shrapnel.	45	1,830	8,500	Wheeled carriage.	8	1,000	Siege operations.
7-inch howitzer	12	4	8	Shell and shrapnel.	105	1,100	8,000	Wheeled carriage.	8	1,000	Siege operations.
7-inch siege mortar...........	7	2	0	Shell and shrapnel.	125	710	4,340	Platform carriage.	....	......	Siege operations.

howitzers of the same mobility firing a projectile weighing 60 pounds; also 4.7-inch rifle firing a projectile weighing 60 pounds, and 6-inch howitzers of the same mobility firing a projectile weighing 120 pounds.

8. This plan has the merit of consistency in that the weights of projectiles are exactly doubled for each increase of caliber. It would appear that the 3.8-inch howitzer and the field gun of the same caliber are relatively the least essential and might be omitted, and the light field gun would be improved by increasing the caliber and the weight of projectile to about 10 pounds, making a more efficient projectile for both shell and shrapnel fire.

6 and 15 pounder guns to give 3,000 f. s. initial velocity and a 6-inch wire-wound gun to give 3,500 or 3,600 f. s. velocity have been projected.

9. The diversity of calibers of guns is remarkable, indicating how completely every phase of the problem of attack or defense is sought to be covered. When we examine further the designs of modern guns, of the carriages and mounts, sometimes operated by electricity, the projectiles, fuses, powders and high explosives, the instruments and methods employed for range finding and for fire control and direction, it is apparent how complex the science of ordnance has become. Simplicity of means and design is especially desirable in the military service, which requires the greatest perfection in training of personnel. Moreover, the material is through necessity generally designed with high regard for economy in weight and dimensions and withal is subjected to most severe usage. The present period is marked by a strong tendency to increased complexity and expense, engendered by the rivalry of nations and made possible by the advance of knowledge in mechanical appliances. Experience has taught that this tendency is no more to be successfully combated in military than in civil progress, and we must employ the common medium of trial to seek out what is good and reject what is unsatisfactory.

Historical Sketch of the Development of Designs and Trials of Experimental Material.

10. It is to be remarked that the efficiency of the land armament existing at the close of the Civil War has not since been

TABLE II.

SEA COAST ARMAMENT.

Piece.	Length of bore.	Powder charge.		Projectile.		Muzzle velocity.	Muzzle energy.	Estimated perforation of hard faced steel plate.	Approximate effective range.	Character of Mount.	Employment.
				Kind.	Weight.						
	Calibers.	Lbs.	Ozs.		Lbs.	F. s.	Ft. tons.	Inches.	Yds.		
1.45-inch automatic gun.	30	....	1.5	Percussion shell.	1	1,800	....	....	3,000	Light wheeled carriage.	Movable against landing parties.
2.24-inch R. F. gun	50	....	20	Percussion shell.	6	2,400	255	....	5,000	Rampart (wheeled).	Defence of mine fields: against torpedo or other light armored vessels and landing parties.
3-inch R. F. gun.........	50	5	....	Steel shell and shrapnell.	15	2,600	703	4.36	6,000	Balanced pillar and pedestal	
5-inch R. F. M. 1897....	45	16	8	Armor piercing shell (thick and thin wall) and shrapnell.	58	2,600	2,580	7.25	7,000	Balanced pillar and barbette.	Against armor of medium thickness and to combat similar guns on shipboard.
5-inch R. F. M. 1900....	50	26	....		58	3,000	3,430	8.70	7,500		
6-inch R. F. M. 1897....	45	29	12		106	2,600	4,700	9.25	8,000	Disappearing and barbette.	
6-inch R. F. M. 1900....	50	43	....		106	3,000	6,250	11.0	8,500		
8-inch rifle. M. 1888....	32	80	....	Armor piercing steel shell thick and thin wall.	318	2,250	10,000	12.0	10,000	Barbette and disappearing.	Against heavy armored vessels.
10-inch rifle. M. 1888-95..	34	155	....		606	2,300	21,100	16.9	11,000	Barbette and disappearing.	
10-inch rifle. M. 1900	40	245	....		606	2,560	26,120	19.7	12,000	Disappearing.	
12-inch rifle. M. 1888-95..	34	275	....		1,048	2,300	36,670	21.3	12,000	Barbette and disappearing.	
12-inch rifle. M. 1900	40	375	....		1 048	2,560	45,420	24.8	13,000	Disappearing.	
16-inch rifle. M. 1895	35	660	....		2,400	2,300	88,050	32.0	16,000	Disappearing.	
12-inch mortar. M. 1886..	9	44	...	Steel deck piercing and torpedo shell.	800	1,200	7,990	7.2	11,000	Spring return carriages.	Against deck of vessels and for under water torpedo effect.
12-inch mortar. M. 1890..	10	52	2		1,000	1,150	9,170	7.9	11,000		
12-inch mortar. M. 1890..	10	56	8		800	1,325	....	....	12,240		

equalled until the present period. In 1866 the artillery of the land service was deemed very efficient. It comprised, however, only muzzle-loading guns, and the introduction of rifling, with its accompanying change from the round to the elongated projectile and consequent increase of weight of projectile, range and power generally, had not then reached the larger calibers. The principal rifled guns were the 3-inch wrought-iron field rifle, the 4.5-inch cast-iron siege rifle and the Parrott 10-pounder, 30-pounder, 100-pounder, 200-pounder and 300-pounder rifles made of cast-iron reinforced with a coiled and welded wrought-iron band shrunk over the breech. Bronze smooth-bore guns were represented in the Coehorn mortar, mountain howitzers and the well-known 12-pounder Napoleon field gun. Cast-iron smooth-bores constituted the remainder, and comprised 8-inch howitzers and 8 and 10 inch mortars for siege equipment, with 13 and 15 inch mortars and 8, 10, 15 and 20 inch Rodman smooth-bore guns for sea-coast armament. The carriages were of simple pattern. Those for field guns were made with the flasks and trails of wood, and those for the sea-coast guns, which were made of iron, comprised principally a chassis and top carriage. The chassis rested upon wheels upon traverse circles enabling the piece to be moved in azimuth by hand bars inserted in sockets in the traverse wheels. The top carriage was fitted with eccentric wheels at the front to provide rolling friction in part for running into battery and sliding friction for recoil. Data pertaining to some representative pieces of this equipment are as follows:

Piece.	Powder charge. lbs.	Projectile. lbs.	Muzzle velocity. f.s.	Elevation. degrees.	Range yards.
5.82-inch smooth bore mortar	.5	17.75	....	45	1200
3-inch rifle (Parrott)	1.0	10.5	1232	20	5000
4.2-inch rifle (Parrott)	3.5	30.	1293	25	6700
6.4-inch rifle (Parrott)	10.	100.	1300	35	8450
13-inch smooth bore mortar	20.	223.	....	45	4650
20-inch smooth bore gun (Rodman)	200.	1080.	....	25	8000

11. The magnitude of the armament existing at that time may be inferred from the statement that the number of Rodman smooth-bore guns available was about 425 8-inch, 1,000 10-inch, 305 15-inch and 2 20-inch guns.

12. Great merit was attached to the heavy, smooth-bore guns for their " " racking " effect, even after the introduction of

armor on ships. Nevertheless, the advantage of elongated projectiles with their "punching" effect was recognized, and 8 and 12 inch cast-iron rifles were made for trial as early as 1861.

13. The period of 20 years from 1866 to 1886 was marked by the introduction of breech-loading in place of muzzle-loading guns and the gradual substitution, first, of wrought-iron, and afterwards, of steel for cast-iron in their construction. This period may well be styled experimental in this country, except that it resulted in the production of some 210 8-inch muzzle-loading rifles converted from 10-inch Rodman smooth-bore guns by lining them with wrought-iron or steel tubes and a number of 3.2-inch steel breech-loading field guns. The 8-inch converted rifle served a temporary purpose, but the steel field-gun, with its metal carriage (Appendix 26, Report of 1887) using bow spring wheel brakes to restrain the recoil on firing, has with some modifications given good service even as late as the Chinese expedition of 1900. It is now to be replaced, however, by a quick-firing gun with long recoil carriage, model 1902. During this period the total amount expended by the Ordnance Department to include the first cost of experimental guns and of those supplied for service, amounted to somewhat less than $1,500,000.

14. Satisfied apparently by the existing armament of 1866, in connection with the reaction from the expense of the Civil War, it was not until 1872 that Congress made provision to attempt a revision of the armament, when the so-called Heavy Gun Board was appointed. The continued agitation of the subject may be inferred in noting that the Getty Board was appointed by act of Congress, approved March 3, 1881; the Gun Foundry Board by act of Congress March 3, 1883; the Armament Board by act of July 5, 1884; and finally the Endicott Board by act of March 3, 1885. During the same time there were reports from the special Senate committee, Senator Logan chairman, appointed August 2, 1882; the Senate select committee on ordnance and warships, Senator Hawley chairman, appointed July 3, 1884; and a similar House committee, with Mr. Randall chairman, July 6, 1884. The Endicott Board completed its labors in 1886, submitting a scheme of sea-coast armament which was virtually adopted by Congress in the Fortification Act of September 22, 1888, and while modified from time to time as occasion required, is still being carried out. At this time, after a long contest before the committees of Congress

with the advocates of different systems, including the use of cast-iron, it had been demonstrated that the built-up forged-steel gun recommended by the Ordnance Department could be relied upon.

15. A somewhat extended description of the experimental guns of this period and their trials is contained in a paper read before the military service institution at Governor's Island, November 26, 1887, and brief reference only will be made to them here. The Heavy Gun Board of 1872 selected nine systems for trial, to wit:

16. Muzzle-loading guns: (1) Dr. W. E. Woodbridge; (2) Alonzo Hitchcock; (3) Cast-iron guns lined with wrought-iron or steel tubes.

17. Breech-loading guns: (1) Frederick Krupp; (2) E. A. Sutcliffe; (3) Nathan Thompson; (4) French and Swedish systems.

18. Miscellaneous: (1) H. F. Mann; (2) Lyman multicharge gun.

19. The Fortifications Bill of 1883, embodying the recommendations of the Senate ordnance committee and the Getty Board, authorized the continuation of the conversion of 10-inch smooth-bores into 8-inch muzzle-loading rifles, and in addition the trial of five different systems of gun construction and two types of breech mechanism, as follows:

20. Built-up forged steel breech-loading rifles with slotted screw breech closure.

21. Cast-iron breech-loading rifles.

22. Combined cast-iron and steel built-up breech-loading rifles and rifled mortars of the same system with slotted screw breech closure.

23. Wire-wound breech-loading rifles.

24. The multi-charge gun.

25. The Mann breech mechanism.

26. The Yates breech mechanism.

27. During the period 1873 to 1882 trials were also made, at Sandy Hook Proving Ground, with breech-loading field guns and the Dean 3.5-inch mandreled bronze gun. The Dean gun was produced in 1877. It was subjected to a firing test which, so far as it went, proved the good quality of the material, but it was a muzzle-loading gun made after a design already out of date and gave inferior ballistic results. This system has been extensively used in Austria, as proposed by General Uchatius, for field and

lighter siege guns, but was not successful in larger calibers. The breech-loading field guns that were tested included the Sutcliffe, Moffat and Krupp breech mechanism. Preference was found in these tests for the Krupp mechanism, but subsequent tests led to the final adoption in our service of the slotted screw breech mechanism.

28. The gun proposed by Alonzo Hitchcock was a 9-inch muzzle-loading rifle designed to be made by butt welding disks or cheeses of wrought-iron and forming a solid piece to be bored out for the gun. After nearly three years of labor the project was abandoned as being too difficult and costly, if not impracticable, to be fulfilled.

29. The Sutcliffe and Thompson guns were cast-iron breech-loading rifles of 9 to 12 inches caliber. There is a general similarity in their breech mechanism in that the block is rolled to one side for inserting the charge. The Sutcliffe gun was fired in all 26 rounds and the Thompson 2 rounds, when the guns were sent to the Philadelphia Centennial Exposition and not afterwards tested.

30. The principal feature of the multicharge gun consists in utilizing the accelerating principle for the action of the powder upon the projectile. This is sought to be obtained by having a series of powder charges placed in pockets at intervals along the bore near the breech which are ignited by the inflamed gases of the breech charge following up the passage of the projectile over the opening of each powder pocket into the bore. The gun was patented by A. S. Lyman, who, in connection with J. R. Haskell, began experimenting to test the system about 1855. Guns of 2½-inch, 6-inch and 8-inch caliber have been tested. The last test was that of the 8-inch steel gun, with one breech and two auxiliary powder chambers, at Sandy Hook in 1897. At the second round the metal between the forward chamber and the bore was crushed in, due probably to premature ignition of the powder charge in that pocket by passage of the gas in advance of the projectile.

31. The principle involved in the Mann breech mechanism is to completely separate the longitudinal from the tangential strains due to firing a gun. This is accomplished by making the breech of the gun separate from the gun body and connecting it by heavy side straps with the trunnion supports on the carriage. Several calibers of this gun were tested between 1862 and 1884.

The last occasion was a 6.5-inch gun, fired at Sandy Hook in 1884, which was burst at the 24th round.

32. The Yates breech mechanism comprises a couple of concave clamps which open outwards from the breech, and when closed embrace the breech of the gun exteriorly and support a solid head gas check or cartridge case for sealing the escape of gas. The 8-inch rifle was tested at Sandy Hook, 1885-6. The gun was fired in all 312 rounds, when it was destroyed by bursting through the body.

Cast-Iron Rifles.

33. In company with the manufacture of the Rodman smoothbore gun the manufacture of cast-iron in this country was brought to a high state of perfection and exhaustive attempts were made to utilize this metal in the construction of heavy rifled guns. Seven muzzle-loading cast-iron Rodman rifles of 8, 10 and 12 inch caliber were produced between 1861 and 1869. When, however, the 12-inch rifle of 1868 was burst at the 27th round, in 1871, the Ordnance Department recommended that no cast-iron rifles be made for service. Efforts were not, however, relinquished from other sources. Mr. Norman Wiard, in the Nut Island experiments, 1873 to 1875, attempted to show the utility of cast-iron rifles using mitten or subcaliber projectiles but without any reasonable success. Congress subsequently required the manufacture of a 12-inch cast-iron breech-loading rifle in 1883. It was made with 28 calibers length of bore and gave a muzzle velocity of about 1,750 f. s. with 800-pound projectile, corresponding to 17,000 foot tons muzzle energy. After firing 137 rounds the trials were suspended, due to the erosion of the bore which increased rapidly toward the end of the test and became so serious as to lead to the conclusion that it would be unsafe to continue the firing. Again, in 1889, pursuant to act of Congress, September 22, 1888, the South Boston Iron Works submitted a 12-inch cast-iron rifled mortar, which burst explosively on trial at the 20th round.

Converted Guns.

34. A quantity of serviceable muzzle-loading guns dating from 1874, although of low power, were produced by lining the

Rodman smooth-bore guns first with coiled and welded wrought-iron and later with steel tubes. In addition to the muzzle-loading guns, several calibres of converted breech-loading guns were constructed and tested, but without ultimate success. In this design a jacket made of a heavy steel piece was screwed into the breech of a smooth-bore gun and adapted to receive the Krupp breech mechanism. The steel forgings for these alterations were procured in England, and chiefly due to the poor quality of the steel the plans were abandoned. This occasioned an unfavorable opinion regarding the use of steel in gun construction. It was difficult for some years afterwards to convince the doubters that there was taking place a great improvement in the quality of steel for guns gained by knowledge and experience in its manufacture. It was equally unfortunate for the Krupp breech mechanism that its first application to large guns in this country was made in this connection. The slotted screw mechanism was applied to the subsequent experimental guns and became the established type in our service.

Combined Cast-Iron and Steel Guns and Mortars.

35. This method of construction, authorized in 1883, was practically forestalled by the advance in steel manufacture before the experimental pieces were completed and tested. Four experimental types were made, including two 12-inch breech-loading rifles and one muzzle-loading and one breech-loading mortar. One of the rifles comprised a cast-iron body lined for about one-half the length of the bore with a steel tube inserted from the breech. The second rifle, in addition to the half-tube lining, was reinforced by a double row of steel hooping on the cast-iron body, extending from the breech to a distance forward of the trunnion band. The two 12-inch rifled mortars were made with a cast-iron body reinforced by a double row of steel hooping. Of these types the breech-loading mortar only survived for service construction, but gave place to the all-steel mortar in the model of 1890. The hooped guns and mortars of this type exemplified the built-up construction by shrinkage and the principles of this method were carefully applied in making them. The manufacture of the 12-inch hooped rifle was preceded by an experimental construction embodying a complete section of the gun through the powder chamber; that is, a compound cylinder

forming a counterpart of the gun section. The section of cast-iron cylinder used in this trial was cut from the body of the gun casting and the steel parts were of similar material to the forgings made for the gun. The objects accomplished by this means were the verification of the shrinkages calculated for the gun and a practical test of the metals on the same scale as the gun itself.

Cast-Steel Guns.

36. Trials of cast-steel guns were authorized by the act of March 3, 1887, and two guns of 6-inch calibre were procured and tested by the Navy Department. One was made of Bessemer steel by the Pittsburg Steel Casting Company, and the other of open-hearth steel by the Standard Steel Casting Company. The processes of treatment of the castings were left in the hands of the manufacturers and have not been published. It is understood that both guns were cast solid and that the Bessemer casting, after having been bored out, was subjected to a process of heating combined with interior cooling to produce a certain degree of initial tension, but the effect of this treatment has not been ascertained. The open-hearth casting is believed to have been simply annealed. In the firing tests, which were made first with reduced charge and then with the normal charge for guns of this caliber, the Bessemer gun burst explosively at the second round. The open-hearth gun was unduly enlarged after firing twelve rounds. The strength of these guns was insufficient for the powder pressure, which amounted to about fifteen tons per square inch. Specimen tests of the Bessemer steel gave an elastic limit varying from 43,000 to 55,000 pounds, and of the open-hearth steel from 30,000 to 40,000 pounds. On the usual basis of estimate for strength of gun cylinders the open-hearth gun might have supported with safety repeated powder pressure not exceeding about ten tons.

37. Another cast-steel gun, of 8-inch calibre, proposed by the well known and esteemed inventor, Dr. Gatling, was procured pursuant to act of Congress, June 6, 1896, and tested in 1899. In the construction of this gun it was attempted to produce initial tension by mandreling the bore while hot. On trial the gun burst explosively at the fifteenth round, after being subjected to pressures not exceeding about 41,000 pounds. One of the broken fragments, comprising half of the powder chamber,

showed along the ruptured surface three cavities extending from one to four inches into the body of the metal.

38. The Bofors Company, in Sweden, makes guns of cast-steel of a first rate quality. It must be understood, however, that these guns are made with tube, jacket and hoops on built-up principle and the separate parts are nickel steel castings treated to improve their quality in essentially the same manner as the forgings used for other steel guns.

Wire Wound Guns.

39. The modern wire-wound gun is a worthy rival of the built-up forged steel gun. The same principles are employed in the construction of both. Dr. W. E. Woodbridge presented the first plan for a wire-wound gun in 1850, when a 2.5-inch gun was made and tested. Three additional Woodbridge guns, of 10-inch caliber, have also been tested. The first, proposed in 1872, was a muzzle-loader comprising a thin steel tube wound with wire and subsequently dipped as a whole into molten solder to fill the interstices between the wires and consolidate the structure. This gun on trial was ruptured longitudinally after a limited number of rounds. The second 10-inch rifle was a breech-loader made of a cast-iron body wrapped with steel wire. The test was concluded in 1892 after firing 161 rounds. In common with other cast-iron bores it suffered great erosion and was laid aside. The third was a steel 10-inch breech-loading rifle with a steel tube, a steel jacket of cold rolled bars or staves made to form a cylinder fitting the tube over about one-half its length from the breech and steel wire wound over the jacket. The test of this gun, in 1894, produced four serious longitudinal cracks in the bore of the tube after firing 23 rounds. These cracks were generally developed in straight lines following the joints of the enveloping stave cylinder.

40. The Brown segmental wire gun is still on trial. This consists essentially of a steel tube enveloped by a segmental jacket which is wrapped on the outside with steel wire. Two guns of 5-inch caliber have been tested; one by firing 192 rounds, in 1894, and the other 300 rounds, in 1899. In the first of these guns the segmental jacket was formed by twelve wedge-shaped staves forming a cylinder around the tube or liner but the latter extended from the breech to a distance in front of the powder

chamber only, leaving the forward portion of the bore to be formed by the segmental cylinder. In the second gun the wedge-shaped segments are replaced by curved segments forming the segmental tube and the lining tube extends throughout the length of the bore. The latter construction is also used in the 10-inch breech-loading rifle now at the proving ground. It is understood that the company has undertaken to submit for trial a 6-inch gun designed to give a muzzle velocity of 3,500 f. s. with 100-pound projectile.

41. The Crozier 10-inch wire-wound gun consists essentially of a cylindrical tube of the usual type wrapped throughout its length with steel wire and an outside jacket utilized to support the breech block and the gun in mounting. This gun was tested by firing 275 rounds, and recommended in 1896—page 320, report of 1896—by the testing board as a suitable gun for service. The construction of a 6-inch gun on the same general plan, which may be developed to give a muzzle velocity of 3,600 f. s., has been recently undertaken.

Types of Gun Carriages.

42. The development of types of gun carriages followed that of the guns. The first improved type of metal field carriage with bow spring brakes was planned by Lieutenant-Colonel A. R. Buffington, of the Ordnance Department, who installed a plant on an economical basis for its manufacture at Springfield Armory, Massachusetts, in 1887. This type is now being replaced by the long recoil, model of 1902.

43. The present model of 5-inch siege carriage dates from 1896. It is a wheeled carriage with the gun mounted rigidly in the trunnion beds at a height sufficient to fire over a high parapet. In firing, the carriage is anchored to a pintle on the siege platform by means of an hydraulic cylinder that absorbs the recoil. Wedges are also placed behind the wheels. A limber is used with the carriage for travelling and the gun is shifted to the travelling beds to distribute the weight on the four wheels. This type of gun and carriage, with 45-pound projectile, is to be replaced by one of 4.7-inch caliber, with 60-pound projectile, the carriage for which will provide for long recoil gun on carriage to give the same steadiness in firing as the model 1902 field gun.

44. The present 7-inch siege howitzer carriage, model 1899, supersedes the first model of 1890. The height of axis of piece in firing position is reduced in the later model and the carriage made about 1,000 pounds lighter, with simpler construction, improved sights and brakes and improved means for shifting the piece for travelling. The general arrangement of the recoil system is the same as in the original carriage. This was designed by Captain William Crozier, Ordnance Department (Notes on Construction of Ordnance No. 57, January 12, 1891), introducing the novel feature at that time of recoil of piece on carriage for mobile artillery. The piece is restrained on the carriage by a pair of hydraulic recoil cylinders and counter recoil springs in addition to the hydraulic cylinder anchored to the pintle of the platform.

45. The 3.6-inch mortar carriage dating from 1890 is a simple rigid type fired from a small wooden platform. To avoid overturning the carriage when fired with full charges at the lower elevations, it is necessary to anchor it with a rope attached to the front of the cheeks and fastened to some anchorage such as a strong stake driven into the ground. The first model of 7-inch mortar carriage dates from 1892 but has been since improved and modified. The piece is restrained in recoil by a pair of hydraulic cylinders and is returned to the firing position by counter recoil springs. Clip circles are attached to the wooden platform to permit of traversing about a fixed position by means of a hand bar engaging in the teeth of a rack cut in the rear clip circle.

46. The carriages for rapid fire guns of the sea-coast armament, which include the guns up to 6-inch caliber, are of barbette and pedestal type, excepting the 6-pounder and also the 6-inch guns on disappearing carriages. A part of the 15-pounder and 5-inch rapid fire guns have the so-called balanced pillar mount which enables the gun to be lowered behind the parapet when not in action and concealing it from view of the enemy. When raised to the firing position, however, and during the whole period of action the protection is no greater than the ordinary barbette mounting.

47. The 6-pounder carriage is a wheeled mount designed in 1896 for the double purpose to use in a fixed position, with anchorage behind a parapet, and for moving into the open for firing upon landing parties. The piece has a short recoil of

gun on carriage and is unsteady in firing even when firmly anchored. In future construction the 6-pounder guns will be given a pedestal mount. The 15-pounder rapid fire gun carriages have been materially modified since the early type of 1897. The balanced pillar construction then adopted for this caliber as well as for 5-inch rapid fire guns is expensive and found not to be as satisfactory in operation as desired. A fixed pedestal mount was substituted later. The model of 1902 15-pounder carriage embodies the latest improvements including particularly gearing for controlling the traverse of the gun and improved arrangements of the telescopic sight mounting.

48. The first 6-inch rapid fire guns were mounted on a Buffington-Crozier disappearing carriage, model 1898. In 1900 an improved model of barbette carriage, with curved shield, was designed as a type for 5-inch and 6-inch rapid fire guns on barbette mounts. A number of these carriages are now under construction. Later a decision was reached to restore the disappearing mount to the 6-inch gun and it has been urged also to likewise mount the 5-inch guns in future to afford the best protection to the gun and mount and also to the firing party.

Sea-coast Barbette Carriages.

49. A limited number of the heavier sea-coast guns are mounted upon open barbette carriages, designed to be ultimately protected by front shields as in the case of the rapid fire guns on barbette mounts. The types of heavy barbette carriages include the 8-inch model 1892, 10-inch model 1893 and 12-inch model 1892, which are similar in their main features and act upon the "gravity return" principle. The rails of the chassis are inclined upward to the rear and the top carriage carrying the gun is mounted thereon with rollers. The recoil is absorbed by hydraulic recoil cylinders and the top carriage returns to the firing position by rolling down the inclined rails of the chassis under the action of gravity. The loading arrangements for these carriages, particularly the chain shot hoist, are slow and cumbersome. The carriages are geared for ready hand power control in traversing and elevation.

Sea-coast Casemate Carriage.

50. A 12-inch minimum port casemate carriage was ordered from Grusonwork, Germany, 1892, and tested at the Sandy Hook Proving Ground (Appendix 26, Report of 1895). The manœuvring of this carriage is controlled in large part by hydraulic machinery, the pressure for which is taken from an accumulator fed by a hand pumping apparatus supplied by six pumps served by twelve men. A steam pump may be substituted for the hand power. The carriage fulfilled the conditions of the trial and was accepted as a type suitable for service. It can be adapted for guns of 8, 10 or 12 inch caliber. However, no call has been made for this type of mounting for guns in our service since no armored casemate forts have been erected.

Mortar Carriages.

51. 12-inch mortar carriages were amongst the first to be furnished in quantity with the new armament. Two type carriages were ordered in 1890, one from Easton & Anderson, of the Raskazoff spring return pattern, and the other from Whitworth, of the Canet pattern. The Easton & Anderson carriage combined hydraulic recoil cylinders with two piles of Belville springs to return the piece to the firing position. The carriage was accepted, with modifications, as a type in 1891 and eighty-five of them have been manufactured for service. Considerable difficulty was met in manufacturing the Belville springs, and although this was overcome by careful investigation and tests made with the Emory testing machine at Watertown Arsenal, it was finally determined to use coiled springs in place of the Belville.

52. The Canet carriage comprised hydraulic cylinders with requisite throttling arrangement to check the recoil and a recuperator holding compressed air under an initial pressure of about 900 pounds per square inch to return the piece to the firing position. The leakage of air from the recuperator during the trial was comparatively slight. The manœuvring, however, was slow and the general results of the trial failed to recommend the type for service.

53. The present type of 12-inch mortar carriage, model 1896, of which about 300 have been provided, was designed by Captain W. B. Gordon, Ordnance Department, U. S. A. The first car-

riage was made at the West Point foundry, Cold Spring, New York, in 1895, and after trial was adopted with the minor modifications shown to be necessary. In this carriage the recoil is checked by hydraulic cylinders and the piece returned to the firing position by springs compressed during recoil. A number of double coil spiral springs are placed under the saddle that supports the piece at a point about one-third its length from the pivot connection of the saddle with the racer of the carriage. The recoil is checked by two hydraulic cylinders with return passage. The cylinders are trunnioned and hung in brackets bolted to the upper surface of the racer, one on each side of the carriage. The lower ends of the cylinders are connected by an equalizing pipe. The carriage is geared for traversing and elevation by hand power. Recent modifications of this carriage have been found necessary to strengthen the racer and other parts to support the strain due to the use of full charges of smokeless powder. Improved quadrants for giving elevations are also under trial.

Gun Lift Carriages.

54. The first form of disappearing carriage to be adopted in the service is the gun lift carriage, which is used in the emplacements of special construction. The gun and carriage complete are lowered in a shaft and completely concealed in the loading position, the loading being done through a gallery opposite which the breech of the gun stands when lowered. The power for raising and lowering is furnished by hydraulic apparatus connected with an accumulator. The carriage proper differs but little from the gravity return barbette carriages, excepting an automatic arrangement of the recoil cylinders to retain the gun in the retracted position for lowering into the shaft and loading. The expense for installation and maintenance of the gun lift battery, the slow rate of fire of the guns and the later adoption of the disappearing carriage has led to a very limited installation of the gun lift carriage proper. A few altered gun lift carriages have been mounted in ordinary barbette emplacements.

Disappearing Carriages.

55. The following disappearing carriages were made and tested prior to or concurrent with the present adopted types of

Buffington-Crozier carriage for 6, 8, 10 and 12 inch high power sea-coast guns, namely:

Pneumatic	10-inch	carriage,	1st	pattern,	produced	in	1891
"	"	"	2nd	"	"	"	1899.
Gordon	"	"	1st	"	"	"	1892.
"	"	"	2nd	"	"	"	1894.
Howell	"	"			"	"	1898.

The first pneumatic carriage was furnished under specifications calling for the piece to be raised to the firing position, its recoil controlled, the carriage traversed and the charge raised and loaded in the gun by compressed air power. Also to provide a hand pump capable of charging the recoil cylinders so that steam power might be dispensed with, and to provide means for traversing, elevating and loading the piece by hand power. This carriage fulfilled the tests prescribed for acceptance, but its slowness in operating by hand power and general performance, coupled with the extent of steam plant required, failed to commend it as a service type. The second carriage failed to fulfil the tests for acceptance.

56. The first Gordon 10-inch carriage comprised essentially a heavy bed plate supporting two side frames with a pivot plate fastened under the bed plate; the whole resting upon a traverse circle, for the substructure. The top carriage carrying the gun is attached to the inner ends of two double crank arms on each side, journaled through the side frames and carrying heavy counterpoise plates outside of the frame. In recoil the crank arms revolve 180 degrees, the gun being lowered at the same time that the weights are raised by revolution of the crank arms. Two hydraulic cylinders are connected with this motion, in which the piston rods are pushed to the front and force liquid into an air chamber to store therein sufficient energy to raise the piece to the firing position by air pressure on opening a valve. On trial the carriage was found to be slow in operation but showed enough merit to warrant the construction of an improved pattern. The trial of the second carriage was fairly satisfactory, and resulted in an opinion from the testing board that the carriage might be used in emplacements where a center pintle carriage furnishing an all-around fire is desired.

57. The Howell 10-inch is a counterpoise carriage embodying the plan in which the gun levers, upon the ends of which the gun is supported, revolve about a fixed axis, and the counter-

weight is supported at the other ends of the arms by interposition of an hydraulic cylinder to reduce the shock of suddenly starting the counterweight. On trial the carriage passed the firing tests prescribed for acceptance; it, however, weighed nearly twice as much as the Buffington-Crozier and did not equal the latter in general performance.

Buffington-Crozier Disappearing Carriages.

58. In 1890 Captain William Crozier (now Chief of Ordnance, U.S.A.) redesigned a type of counterpoise carriage that had been proposed some years previously by General Buffington when a Captain of Ordnance. The principal features of the carriage relating to the disappearing principle comprise a pair of gun lever arms, with the gun supported in trunnions at the upper end and a counterweight attached at the lower end. The counterweight is raised when the gun recoils in firing and becomes operative to return the gun in battery upon releasing the pawls. The gun lever arms are mounted on a horizontal axis, which is journaled in the top carriage. The top carriage moves on rollers to the rear in recoil and carries with it two hydraulic cylinders with throttling bars giving variable openings in the passage of the piston head so designed as to make the resistance uniform throughout the recoil. The counterweight affords some assistance in checking the recoil. During the recoil the trunnions of the gun describe ellipses in passing to the loading position. Sample 8 and 10 inch carriages of this type were tested in 1894 and at once adopted for service. The Ordnance Board, which conducted the tests, reported as to the 8-inch: "The test of this carriage has demonstrated that it possesses in a marked degree the properties which should pertain to a disappearing carriage for high power guns. It is simple in construction so that its parts and purposes are easily understood." Also as to the 10-inch: "The advantages of this system of disappearing carriage, as set forth in the report of the Board on the 8-inch carriage, are confirmed and emphasized by trial of a carriage adapted for a gun of much greater caliber and power, and it is the opinion of the Board that the exhaustive test to which this system has now been subjected demonstrates that on account of the simple features of construction involving no valves, pumps or other complicated appliances, and the fact that by methods easily understood by the average artillery soldier the operations

of loading and manœuvring are effected with remarkable ease, certainty and rapidity, it is worthy of adoption in the service on all sites except those where an all-round traverse is absolutely necessary." The all-around fire model of 10-inch carriage was produced in 1896. Improved models of the 8 and 10 inch carriages for limited field of fire were also produced in 1896, comprising a center turn-table instead of a forward turn-table and rear traverse circle, and live chassis rollers instead of axle rollers. The field of fire was increased from 150 to 170 degrees, and a considerable saving of counterweight effected a material reduction in the effort required to haul a piece down by hand.

59. When the model 1896 12-inch carriage was tested and the time required to fire ten rounds was 16 minutes 57.2 seconds, it was felt that a notable advance in gun carriage construction had been made, since it had previously been considered impracticable to mount guns of this size and power upon disappearing carriages. An improved model of 12-inch carriage was brought out in 1897.

60. The 6-inch disappearing carriage, model 1898, includes a sighting platform with handwheels and gearing that enable the gunner himself to control the direction and elevation of the gun from his position at the telescopic sight.

61. Up to this date the carriages were arranged to be operated in traversing and elevation and for retraction by hand power. They are now being modified to use electric power for these operations. The latest patterns of the Buffington-Crozier disappearing carriage, model of 1901, embody all of the desirable improvements indicated by experience and practice had with the carriages up to the present time, including the application of a system of electric control in connection with an improved method of sighting.

62. The manufacture of sea-coast carriages proceeded rapidly after the types were established. At the close of the fiscal year ended June 30, 1898, 398 carriages had been delivered; in 1899 the number was increased to 605, and in 1903 to 1,000.

Industrial Development Accompanying the Demand for Modern Ordnance.

63. The built-up forged steel gun, as is generally known, is the type of construction used in existing land as well as navy ordnance. Before discussing the principles of construction of

the type, reference will be made to the industrial development in this country, which may be said to be a direct consequence of the demonstrated success of this type of gun in the early eighties. That is to say, progress in armament was waiting for a good gun and after its adoption the rest followed as a matter of course. The immense operations of the Navy Department will not, of course, be forgotten by any one in this connection, but these remarks must be confined to the influence of the War Department, and more particularly the Ordnance Bureau of the War Department, which is charged with the procurement of the land ordnance. The first active steps for manufacture of high power steel guns were taken in 1883, when a field gun was ordered and the forgings procured for 8-inch and 10-inch sea-coast rifles. The 8-inch rifle was completed in 1886, at the West Point foundry, where I was at the time on duty as inspector of ordnance. Two years later, by the act of September 22, 1888, a long period of inaction was terminated and Congress made liberal appropriations for the manufacture of these guns, and each year since has continued appropriations enabling rapid progress to be made towards the completion of the plans for sea-coast defense proposed by the Endicott Board in 1886, as well as for improving the field and siege armament. The progress shown in the procurement of sea-coast armament dates, therefore, from 1888.

64. In his annual report of 1903 the Chief of Ordnance, General Crozier, gives the number of guns manufactured which have been issued or are available for issue to fortifications, together with nearly an equal number of carriages, as follows:

6-pounder rapid-fire guns	70
15-pounder rapid-fire guns	119
4-inch rapid-fire guns	4
4.7-inch rapid-fire Armstrong guns	34
5-inch rapid-fire (Ordnance Department) guns	32
6-inch rapid-fire Armstrong guns	8
6-inch rapid-fire (Ordnance Department) guns	42
8-inch breech-loading rifles	85
10-inch breech-loading rifles	134
12-inch breech-loading rifles	127
12-inch breech-loading mortars	371
Total	1,026

65. The report of the Secretary of War for the same year states that the provision of heavy guns in sea-coast fortifications

is far advanced. The most pressing requirements in the way of material are a further provision of rapid fire guns and the installation of fire control apparatus. A table is given showing that about $51,342,800 have been expended for armament, including material in process of manufacture, and the amount remaining to be appropriated for armament to complete the project is $12,111,775.

66. In the inception of this work the Chief of Ordnance, General Benét, on April 3, 1883, addressed a circular letter (report of 1883, page 6) to more than twenty of the principal steel makers in the United States to ascertain existing facilities for making gun steel forgings and inviting an expression of their views on the subject. The letter stated the physical qualities required in forgings and summarized the methods of manufacture in vogue in other countries so far as known at the time. The replies to this letter were, with one or two exceptions, adverse to taking up the work and introducing new plants, being no doubt largely influenced by the uncertainty at that time attending its future. The Midvale Steel Company was the first to respond and undertake the manufacture, first of plain hoop and then of trunnion hoop forgings, and by improving its facilities was able in 1885 to accept an order for a complete set of forgings, tube, jacket and hoops for an 8-inch rifle. Meantime, in the absence of facilities in this country for making the larger forgings, in all six gun tubes, three jackets and five trunnion hoop forgings for different types of experimental 8-inch, 10-inch and 12-inch guns were ordered from abroad from Sir Joseph Whitworth & Company and the Creussot Steel Works. Meantime also, the Ordnance Department continued with limited means to procure forgings for field and siege guns and hoop forgings for sea-coast guns—the latter largely for experimental purposes, which produced results of great benefit to the art of manufacture and knowledge of gun construction. The Cambria Iron Works undertook a part of these orders but did not materially enlarge its plant. The net result was that when in 1888 through act of Congress the Department was able to give larger orders for forgings, the Midvale Steel Company and the Bethlehem Iron Works were prepared to fill them for all the sizes of forgings demanded. Both companies received remunerative orders and that of the Bethlehem company included twenty-three 8-inch, twenty-three 10-inch and fifteen

12-inch sets of complete forgings for guns. Up to June 30, 1903, an approximate estimate of the amount of gun steel forgings procured by the War Department since 1888 from domestic manufacturers is 25,700 long tons, at an average cost of about $500 per ton, or an aggregate of about $14,000,000.

67. I think it may be fairly claimed that the thorough and systematic course of investigation of quality and manufacture pursued by the Ordnance Department in the incipiency of gun steel manufacture in this country was most influential in bringing about a speedy and satisfactory state of manufacture and in making it profitable as well by establishing the excellence of the built-up forged steel gun. One of the first steps taken was to test the efficiency of oil tempering as compared with simple annealing. Two hoops of open-hearth steel, about 45 inches interior diameter and 4 inches thick, were procured from the Midvale company, one having been annealed, oil tempered and again annealed, and the other simply annealed. Test specimens were taken from the hoops and the hoops themselves were assembled with relatively heavy shrinkage upon cast-iron cylinders. The results (Notes on the Construction of Ordnance, No. 25) were, first, to establish the superiority of the oil tempered and annealed metal on account of its high elastic limit and great extensibility within that limit, and, second—which was highly important—to establish a striking similarity between the behavior of the metal in the specimen tests and that of the hoops as a whole in the shrinkage tests. This, indeed, gave a basis for all future shrinkage work, since it is upon the tests of detached specimens that we must judge generally of the physical qualities of the metal.

68. Next followed shrinkage and specimen tests of trunnion hoops to determine the quality of metal to be obtained in these irregular pieces. And then the actual construction of a type sea-coast gun was preceded by building up a compound cylinder made to be a complete counterpart of the gun in the section through the reinforce. The purposes of these experimental constructions were fully realized, namely: to obtain such data as could be made available in future construction of the guns; to determine the behavior of the elementary cylinders in combination under the theoretical shrinkages previously deduced by a mathematical application of the formulæ and thus test the theories upon which the formulæ are based; to observe the in-

dividual behavior of the elementary cylinders; and, finally, to determine whether the shrinkages so deduced should be applied in the future construction of the guns or to what extent they should be modified for that construction.

69. An interesting discussion of the quality of steel most suitable for guns is contained in the proceedings of the Naval Institute No. 40, 1887. The effort was made to show that the steel which the Army and Navy Ordnance Bureaus were then advocating was of a grade of so-called high steel with uncertain strength and properties and that guns should be made of mild steel with little or no carbon and having an ultimate strength of from 55,000 to 65,000 pounds. The commercial advantage claimed was that this grade of steel could then be readily manufactured in our own country and a high price would necessarily be paid for the special steel which the government was demanding. The discussion was convincing in showing that what was plainly wanted was a metal possessing a high elastic limit with good ductility and ultimate strength to withstand the pressure to which a gun may be subjected without deformation. Events have since shown that the constant demands of the government for maintaining a high standard of quality and to improve it wherever practicable have been of great benefit to the domestic trade since the qualities of high grade steel manufactured for all purposes in this country are not surpassed elsewhere.

70. The physical qualities prescribed for forgings for guns of 8 and 12 inch caliber and upwards are:

	Elastic limit pounds per sq. in.	Tensile strength pounds per sq. in.	Elongation after rupture per cent.	Contraction of area per cent.
Tube	46,000	86,000	17	30
Jacket	48,000	90,000	16	27
Trunnion hoops	50,000	90,000	14	20
Cylindrical hoops	53,000	93,000	14	20

71. The steel is ordinarily made by the open-hearth process with a medium per cent., about 45, of carbon. Steel containing about $3\frac{1}{2}$ per cent. of nickel is now required for breech blocks and spindles with an elastic limit of 70,000 pounds. It is also supplied for field guns and the tubes and jackets of 5 and 6 inch rapid fire guns, the prescribed limit being 65,000 pounds. The tube and jacket of the 16-inch type gun were made of nickel steel, since the manufacturers would not otherwise guarantee the

required elastic limit of 60,000 pounds in the metal. The cost of nickel steel for the present acts as a bar to its more extended use. The gain in elastic limit is about 30 per cent. over previous standards.

72. The establishment of the army gun factory at Watervliet Arsenal was authorized by the act of September 22, 1888. The tools and equipments were purchased and sufficient machinery installed in the new building to begin operations in September, 1890, or within twenty-one months from the time ground was first broken for building. Up to June 30, 1903, this factory, besides a number of field and siege guns and 12-inch mortars, had completed fifty-seven 8-inch, one hundred 10-inch, one hundred and twenty-five 12-inch and one 16-inch sea-coast guns with forgings furnished by private steel makers. The present capacity of the shops per annum on a basis of 8 hours' work per day is equivalent to:

Field guns, 170 ; siege guns, 10 ; siege Howitzers, 10 ; siege mortars, 11 ; 5-inch rapid-fire guns, 10 ; 6-inch rapid-fire guns, 13 ; 10-inch guns, 16 ; 12-inch guns, 16; 12-inch mortars, 20;

or a total of about 276 pieces of the different calibers. While a large proportion of the guns have been made at Watervliet Arsenal a goodly number have also been made at private factories. All of the 6 and 15 pounder guns have been so procured, as well as a number of sea-coast guns and a large part of the mortars. In 1891 the Bethlehem Steel Company, then under the management of Mr. Frick, made contract with the Department to furnish twenty-five 8-inch, fifty 10-inch and twenty-five 12-inch guns complete and at once proceeded to erect the fine gun finishing plant which to-day distinguishes that company as the most extensive, complete and best equipped private factory for the manufacture of ordnance in the country.

73. The Watertown Arsenal is well equipped for the manufacture of the modern sea-coast carriages by the government. Its capacity is, however, limited and most of the sea-coast carriages are procured from private establishments. The Rock Island Arsenal is well equipped for making field and siege carriages.

74. The Department has no plant for the manufacture of powders or steel projectiles. These constitute a large item of expenditure hardly less in amount than for the guns or carriages.

All material procured from private factories is made after specifications prepared by the Ordnance Department, and under the inspection of its officers stationed at the place of manufacture. It is besides subjected to proof firing tests for acceptance. By this system, coupled with the work turned out from the government shops, the Ordnance Department is admittedly, and justly so I believe, credited with maintaining a high standard of excellence for its products.

Experiments Made to Determine the Utility of Initial Tension and Shrinkage in Gun Construction, Compared with Theory.

75. When a simple, homogeneous cylinder is subjected to interior pressure the surface of the bore sustains the greatest strain tangentially, while the metal of the wall is strained to a less degree depending upon its distance from the axis. The elastic resistance of the cylinder is therefore measured by an interior pressure that will expand the bore to the point where the elastic extensibility of the metal is reached. In a cylinder having initial tension produced by interior cooling or by the shrinkage of cylinders one upon another, or by wire winding, in either case producing a compression of the metal at the surface of the bore, the same law of strains due to an applied interior pressure holds good as in a simple cylinder, but the outer layers of metal having been initially strained will be worked at greater tension than in the case of a simple cylinder and the aggregate resistance to interior pressure will be increased. It is necessary, of course, to observe that none of the outer layers, or indeed any of the metal in the wall, be strained beyond the elastic limit, but the maximum resistance of the cylinder will evidently be attained if each of the indefinably thin concentric cylindrical lamina into which the wall of the cylinder may be conceived to be divided be strained simultaneously to the elastic limit. The object of initial tension or shrinkage is therefore to derive the use of that portion of the elastic extensibility and work in the outer parts of the wall in resisting interior pressure which are not utilized in the simple cylinder. On the other hand the surface of the bore should not be initially compressed beyond the elastic limit of the metal. This is the simple theory involved in the tangential strength of the built-up gun. The necessary longitudinal strength is readily secured.

76. The outer cylinders are to be assembled with such shrinkages that conjointly they will compress the bore of the tube to its elastic limit and yet retain sufficient residual elastic extensibility in each, so that at the moment when the bore of the tube is extended to its elastic limit by interior pressure, the bore of each enveloping will also be extended to its elastic limit. The difference between the resistance of a cylinder with initial tension produced by interior cooling and that of one formed by the shrinkage of cylinders of considerable thickness one upon another here becomes apparent. In the first case the maximum resistance of the whole thickness of wall or the aggregate from the infinite number of cylindrical lamina into which it can be conceived to be divided may be utilized. In the second case the full elastic limit can be realized only at the interior surface of each of the cylinders shrunk on so that the maximum elastic extensibility of the whole thickness of wall is not utilized and there is some loss. In practice, however, since the tube cannot be initially compressed in the state of rest or extended in the state of action beyond the elastic limit of its metal it results that where the thickness of wall is sufficient to divide into three or four layers little or no tangential resistance is lost in the built-up cylinder construction as compared with initial tension produced by interior cooling. These conditions are present in the reinforced parts of the large caliber sea-coast guns, since every gun must have a thickness of wall proportioned to its caliber, and hence the absolute thickness of the wall in the large guns is relatively great.

77. In wire-wound guns the lining tube imposes practically the same restrictions as in the built-up forged steel gun, that is to say, the maximum resistance is limited by the tube which may not be worked beyond its limit of compression at rest or beyond its limit of extension in action.

78. In a solid walled gun properly strengthened by interior cooling it is possible to secure the greatest economy in weight of metal to produce a cylinder of given resistance to interior pressure below a certain thickness of wall in calibers.

79. The following formulæ express the maximum tangential resistance of a simple cylinder or gun and of a compound cylinder, as exemplified in the built-up forged steel, and the wire-wound guns with lining tube.

P is the pressure in pounds per square inch, θ the elastic limit

for extension and ρ the elastic limit for compression of the tube metal in pounds per square inch; R_0 and R_1 stand for the radius of the bore and the exterior of the gun.

$$\text{Simple gun} \quad P = \frac{3(R_1^2 - R_0^2)}{4R_1^2 + 2R_0^2}\theta$$

$$\text{Compound gun} \quad P = \frac{3(R_1^2 - R_0^2)}{4R_1^2 + 2R_0^2}(\theta + \rho)$$

θ and ρ are usually taken as equal, whence it appears that the compound gun will withstand twice as much interior pressure as a simple gun of the same dimensions and quality of metal, provided the tube in the compound gun is worked between the full limits of compression and extension.

80. By plotting curves from these formulæ with abscissæ representing the thickness (R_1—R_0) of the wall in calibers, and with ordinates representing the fractional coefficients in the formulæ, as shown in Fig. 98, it is seen that the resistance to interior pressure increases with the thickness of wall, but the increments are very small beyond a thickness of 1 to 1.25 calibers. The limiting values of P for an infinite thickness of wall, making $R_1 = \infty R_0$, become 0.750 for the simple gun and 1.50 for the compound gun.

81. The shrinkage formulæ that have been applied in the construction of our built-up forged steel guns are published in revised form in Notes on the Construction of Ordnance No. 59, dated July 21, 1891.

These formulæ deal directly with the strains produced in the metal by all the extraneous forces, and by an evaluation of these strains we are enabled to conserve the primary condition that no part of the gun should be subjected to strain beyond the elastic limit of the metal. By modifying Clavarino's formula in relation to longitudinal stress, a set of relations was secured which produced a most satisfactory agreement in actual construction. The formulæ were placed in form to discuss fully the forces engendered in building up a gun with cylinders applied by shrinkage, and thus enabled the fidelity of work to be verified not only in the completed gun but also in the various stages of construction, by measuring the bore of tube and the diameters of successive cylinders when applied.

82. Another valuable attribute of formulas applying directly

PLATE 1, ELASTIC RESISTANCE OF GUNS.

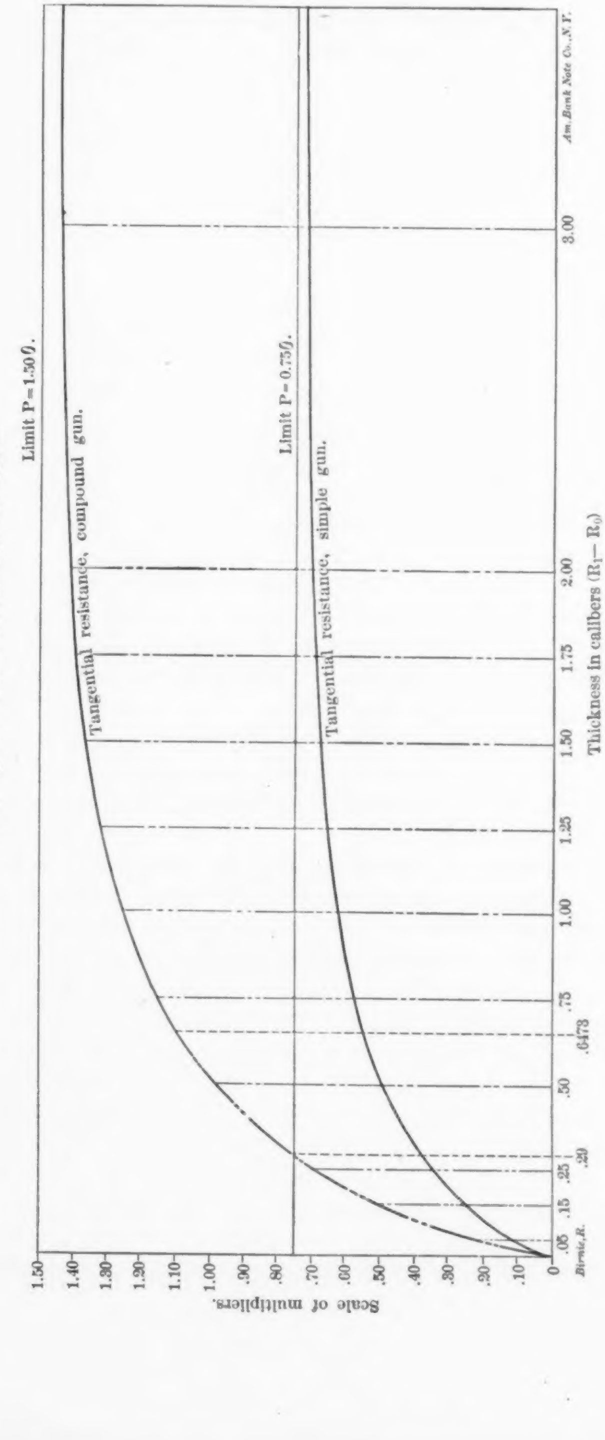


FIG. 98.

to the state of rest of the system is the facility afforded for the adjustment of shrinkages in different sections throughout the length of the gun. In making the computations for all parts of a complete gun, the changes in the number of layers and in sectional dimensions from part to part render it necessary to divide the whole length into a number of sections and compute the resistance, shrinkages and strains for each. These sections cannot, however, be considered wholly independent, as that would give rise to a variety of values which would be inconvenient to apply in practice and would cause undesirable inequalities in strains in passing from one section of the gun to another. Whilst not overlooking the primary consideration to preserve wherever practicable the maximum resistance and in all cases a sufficient elastic resistance to withstand anticipated pressures, two general rules are to be observed, namely: 1. To apply as far as practicable, uniform values for the shrinkages in contiguous sections where the shrinkage diameters are the same, or nearly so. 2. To so modulate the curves of compression or contraction of bore in the state of rest that the final curve will present a comparatively smooth contour, conforming in general to the curve of powder pressure and having no abrupt change of ordinates. As it will in general be most necessary to preserve the maximum resistance in the section of the powder chamber, that section should be considered first and the values belonging to it taken to govern others. Under these considerations it frequently becomes necessary for a given section of the gun to assume certain values, as, for instance, one or more of the shrinkages, the contraction of bore or other conditions and combine the various equations or transform them to obtain desired results.

83. The method of division into sections is illustrated in the figure, Fig. 99, for the 8-inch experimental rifle finished in 1886.* The radial changes in dimensions of the cylinders due to assemblage under shrinkage and the magnitude of the stresses,

---

* The numerous and very short hoops used in this gun were soon changed in future constructions when it was found that the steel makers could readily produce longer hoops. In the 12-inch rifle, model 1900, which is 42 feet in length, there are but two hoops in the "C" row extending from the front end of the jacket to the muzzle of the gun, each of which is about 138 inches long. Similarly the "D" row is formed of a single hoop 108 inches long, and the "A" row of two hoops only, 91, and 109 inches long; the single hoop over the breech in the "B" row is 108 inches long, and the trunnion hoop 28 inches.

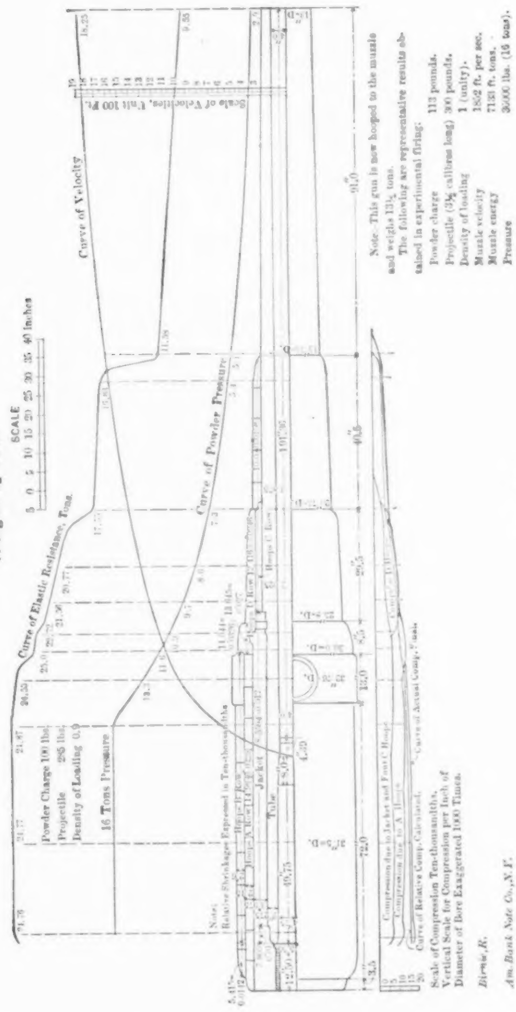


Fig. 99.

in a sectional view, are illustrated in the two figures, Fig. 100, for 12-inch rifle.

84. Practice has shown a remarkable conformity of results obtained in the actual manufacture of guns with those indicated by the formulas applied in their construction. One of the earlier experiments consisted in building a section of gun with 8-inch bore, composed of four cylinders, shrunk one upon the other. The displacements of diameters and lengths caused by each

shrinkage were carefully measured and showed in every instance close agreement with values deduced by the formulæ. For example, the anticipated total compression of the 8-inch bore was 0.0129 inch, and its measured compression 0.0131 inch. The anticipated extension of the exterior of the fourth layer or outer hoop (diameter, 31.5 inches) was 0.0285 inch and its meas-

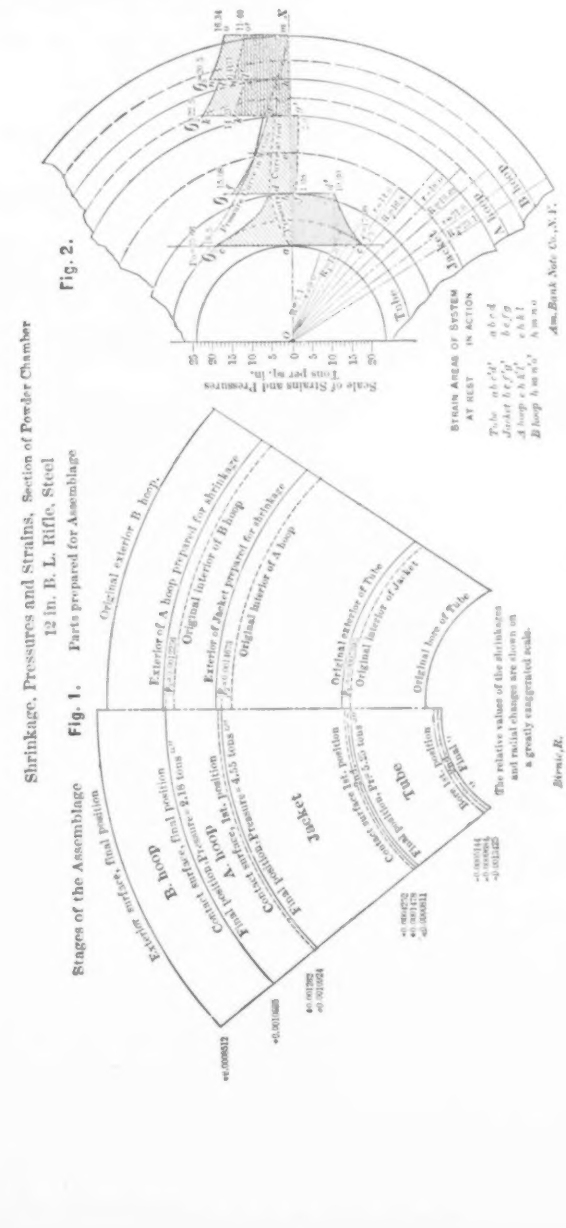


FIG. 100.

ured extension 0.0276 inch. These culminating measurements of the series thus showed a difference of less than 2 per cent. between the actual and anticipated results.

85. In the 8-inch experimental rifle the anticipated compression of the bore in the section of the powder chamber was .0154 and the measured compression .0156 inch. This gun was first made without chase hooping. After firing 24 rounds the bore of the tube at some 15 inches from the muzzle was found to be

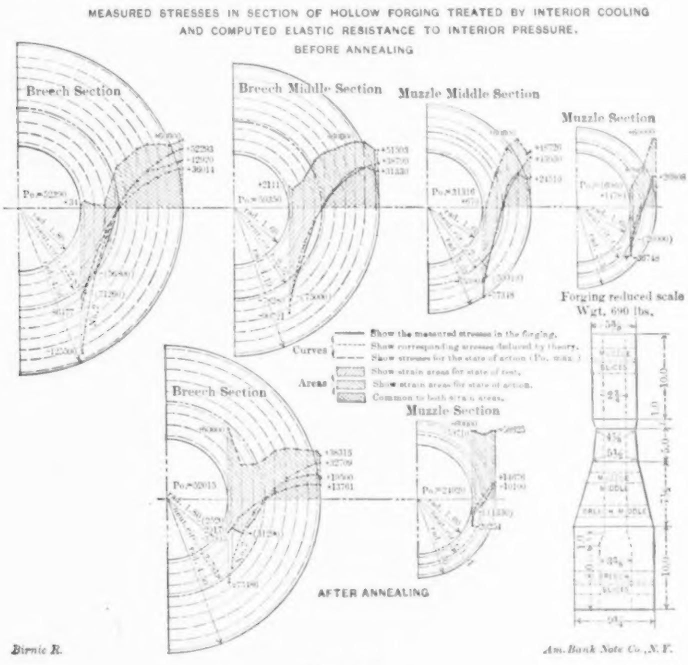


FIG. 161.

enlarged .006 of an inch. It was then decided to return the gun to the factory and extend the hooping to the muzzle. On turning off the chase metal for hooping the bore contracted when metal was turned off showing that there existed a zone of compressed metal at the exterior of the tube damaging to its strength and due to the imperfect treatment at that time in vogue at the Whitworth works where the tube was procured. The chase hooping fully re-established the strength of the gun, which has been fired in all 335 rounds and is still in serviceable condition

PLATE 5. INITIAL TENSION IN HOLLOW CYLINDER.

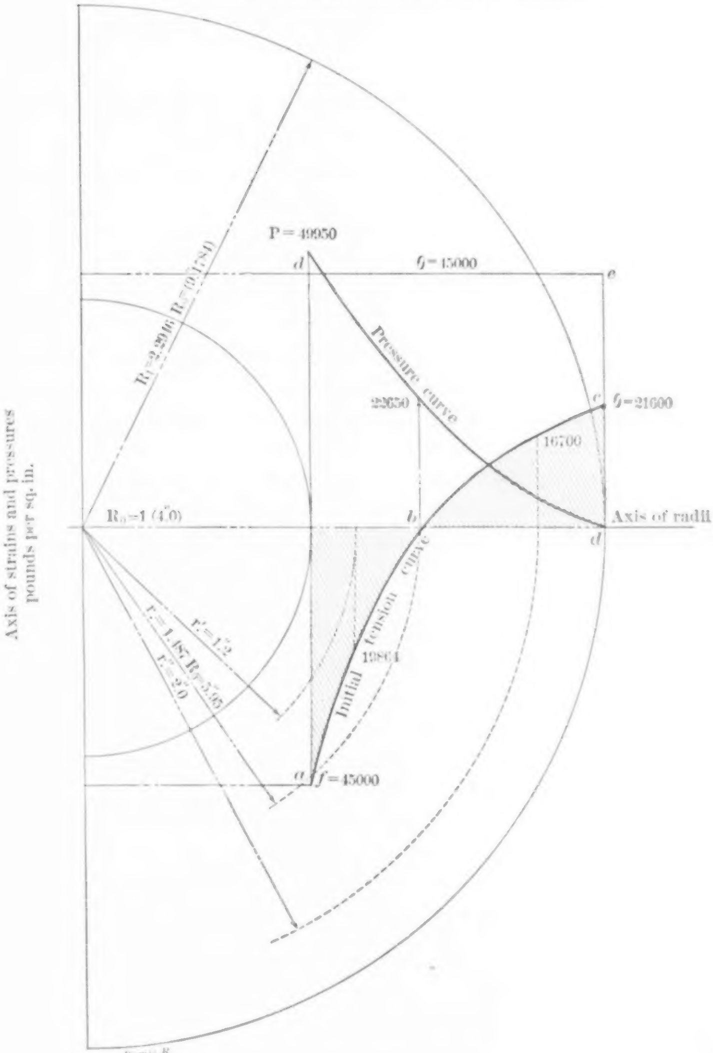


FIG. 102.

excepting erosion of the bore, which detracts from its accuracy.

86. Fig. 101, with figures showing the measured stresses in a forging representing sections of a field gun which was treated by interior cooling for experiment, is taken from a paper read to

the Philosophical Society of Washington, D. C., May 11, 1895, and printed in Bulletin Vol. XIII, pp. 27-102. The wall of the forging was cut into thin concentric rings on four several sections, and the changes measured on the diameter of each ring due to its release from the forging, these changes giving a measure of the strains induced by the treatment.

87. An analysis of the laws governing the resistance of a cylinder with a properly regulated degree of initial tension produced by interior cooling shows that a thickness of 0.65 of a caliber, nearly, will give the strongest tube that can be made from a given weight of metal of given quality. For this thickness in calibers only will all the fibres of the metal throughout the thickness of the wall reach the elastic limit from extension simultaneously under the action of interior pressure. This particular case is illustrated in Fig. 102. The interior cooling being supposed perfectly conducted, the metal at the surface of the bore is compressed to its elastic limit, $\rho = 45{,}000$ pounds assumed. The curve of initial tension is a continuous one extending through the neutral point of stress "b" to a tension of 21,600 at "c," on the outer surface of the cylinder. If now this cylinder be subjected to an interior pressure the stress upon the metal throughout the thickness of the wall will be developed in resisting the pressure until an interior pressure of $P = 49{,}950$ pounds per square inch is reached. At this stage of the pressure the stress curve, which was originally represented by the initial tension curve, a, b, c, is now formed in the straight line, d, e; that is to say, the stress upon the metal throughout the thickness of the wall is at every point equal to the elastic limit of the metal. The elastic tangential resistance of the cylinder to interior pressure is 1.11 times the elastic limit of the metal to be derived from tests of specimens although the wall of the cylinder is only 0.6473 of a caliber thick or actually $8 \times 0.6473 = 5.18$ inches. The same relations would, of course, hold good for a tube with any given diameter of bore and metal of given elastic limit.

88. The following conclusions may here be quoted from the bulletin of the Philosophical Society previously cited.

For cylinders of greater thickness than 0.65 caliber, a state of uniform strain in the wall will be reached in action and passed before the elastic limit of the metal is attained, and with increasing pressure this limit will be fully reached only at the surface of the bore, thus determining the limit of pressure. For such

cylinders the best condition of resistance will be obtained by utilizing the full limit of compression of the metal in the initial tension.

But for cylinders of less thickness than 0.65 caliber, a state of uniform strain in action equal to the elastic limit of the metal can be attained with a compression of bore less than the limit ρ. The thinner the cylinder the less should be the initial compression imposed. It follows that the possible maximum resistance of such cylinders will be obtained by adjusting the initial compression within limits. If the full limit of initial compression were given, the elastic limit of the metal would be reached in action at the exterior of the cylinder sooner than at the bore.

As a consequence, also, of the preceding, the resistance of the cylinders of less thickness than 0.65 caliber, treated by interior cooling, should be directly proportional to the thickness. This treatment gives the means of imparting the greatest resistance so far known to such cylinders.

89. A 3-inch and a 5-inch gun made of single forgings strengthened by interior cooling have been constructed and tested extensively with very satisfactory results. A number of the tubes for 5-inch rapid fire guns have also been made in this way. The outcome of the process in these tubes has not been encouraging for the continuation of this method in view of the trouble experienced and the number of re-treatments necessary to obtain the degree of initial tension desired. Care and experience in manufacture should be able to control this method and apply it advantageously at least to guns of small and medium caliber. The difficulty to be anticipated with guns of large caliber is in procuring a forging for the gun body in one piece of uniform quality. This is, however, also a question of experience and advancement in knowledge of the art, and quite within the limits of future possibility.

The Modern Field Gun (Figs. 103 and 104).*

90. The chief object of improvement in the modern field gun is to obtain the greatest possible rapidity of aimed fire, and this has been rendered possible by the introduction of smokeless powder, since with the front of the battery covered by a heavy cloud of smoke, as formerly, such rapidity of fire as is now attained would not have been possible. Improvements for this purpose have been made in the breech mechanism of the gun, in

* It is regretted that photographs of the model 1902 field gun and limber are not available at this time. These plates show the material tested in 1901-2 which resembles the adopted type.

Fig. 103.—The Modern Field Gun.

the ammunition, and especially in the carriage. Improvements in the mechanism consist principally in a single motion in opening and closing the block, safety devices to prevent firing until the block is fully closed and locked, and eccentric position of the firing pin in the block to prevent premature discharge of the primer in the fixed ammunition when the block is closed.

91. Improvements in the carriage constitute essentially a con-

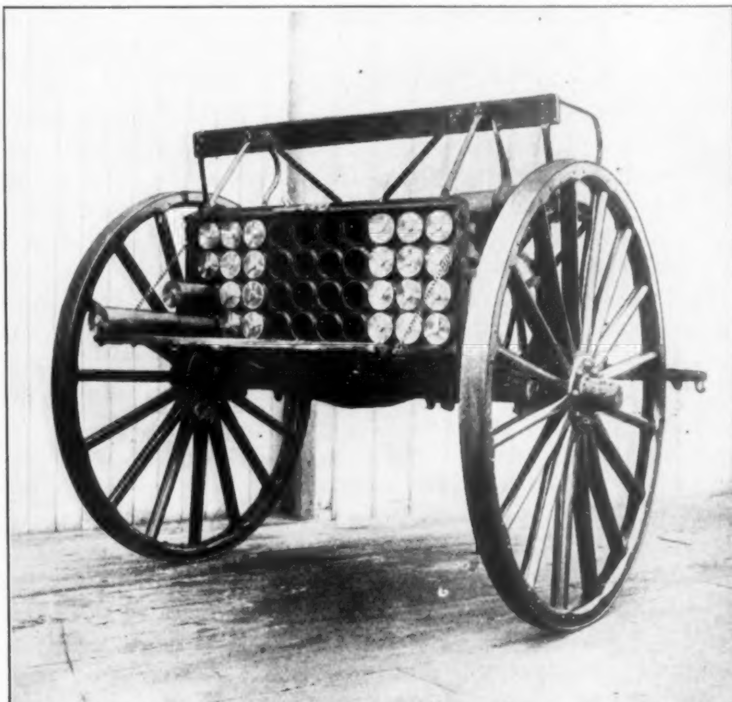


FIG. 104.—LIMBER FOR FIELD GUN.

struction which admits of the gun being fired without recoil of the carriage on the ground, without jump of the wheels, with practically no disturbance of the aim, together with such an attachment of sights that the sighting can be continuously maintained while the gun is being loaded. Improvements in the ammunition are a change from separate loading of the projectile and powder charge to fixed ammunition.

92. In the pattern of equipment now to be replaced the car-

riage recoiled at each round and had to be run to the front, repointed by shifting the trail, and finally the cannoneers must get clear of the carriage to allow the gun to be fired. All of these were time-consuming elements.

93. Three experimental carriages embodying recoil of the gun on the carriage have been tried precedent to the present adopted type, model 1902. First the model of 1898, with the short recoil of gun on carriage, and having an axle traverse to give motion to the gun in azimuth without shifting the trail; next, another short recoil carriage made in 1900. The distinctive features of this were hydraulic recoil cylinders with counter recoil springs in the cylinder, azimuth motion of the gun on the carriage without shifting the trail, provided by a turn-table, and a folding spade at the rear end of the trail. Next came the long recoil carriage allowing a recoil of 44 inches to the gun on the carriage, made with two hydraulic recoil check cylinders and return springs in the cylinder. The two last-named types were entered in the exhaustive tests of field material made in 1901 and 1902, together with several types of foreign and domestic manufacture. The results of those tests led to a combination of good points found, which have been introduced in the model of 1902.

94. The gun is made of nickel steel built up with clips that slide in and secure it to the guide rails of the cradle. A lug under the breech connects the gun with the recoil cylinder. The breech mechanism is of the interrupted screw type and opens with a single motion. The block is bored out to receive the firing mechanism which is placed eccentrically in the block so that in closing, the firing pin will not be brought opposite the primer in the cartridge case until the block is rotated and locked. A safety device prevents firing until the breech is closed. The gun may be fired by lanyard from the rear, but is habitually fired by the cannoneer seated on the right through a firing shaft attached to the carriage. This shaft is so arranged that it cannot be operated until the gun has returned to the safe position in the battery. The advantages of this breech mechanism are rapidity of fire, great power of extraction and ejection of cartridge case, ease of loading in that the cartridge is not required to be pushed into place by hand as in sliding block mechanisms, but the last part of the motion is given by the closing of the block. The mechanism is well arranged for protection of the

parts from dust and injury, especially blowbacks, is simple and no tools are required for dismantling it.

95. The projectiles used are shrapnel and high explosive shell weighing 15 pounds. The gun gives a muzzle velocity of 1,700 foot-seconds, and a range of 6,250 yards with 15 degrees elevation. A somewhat greater range can be obtained by sinking the trail. The maximum range is about 7,500 yards. The accuracy of the gun is excellent at 6,000 yards range, and compares favorably at this range with the 3.2-inch gun, which it replaces, at 4,000 yards range.

The carriage comprises the lower carriage, consisting of wheels axle and trail, together with a rocker and cradle which contains the recoil system and in which the gun slides. The cradle rests on the rocker and is controlled by the elevating gear. The rocker forms a table also in which the cradle can be moved in azimuth 4 degrees on either side of the centre to change the direction of the gun by that amount without moving the trail. The spade at the end of the trail is fixed in position and has wide flanges to prevent sinking in the ground. The recoil equipment comprises the cylinder, piston rod, counter recoil buffer and springs. The cylinder lies in the cradle and is surrounded by the springs, and its rear end is attached to the breech of the gun. The recoil springs are made of thin steel riband rectangular in shape coiled on edge. The column of springs comprises three similar coils assembled with tension sufficient to return the piece when fired at 15 degrees elevation. The piston rod is bored out for the throttling bar and the interior of the cylinder has three ribs of variable profile to control the recoil by openings through which the liquid in the cylinder passes. The variable openings in the cylinder are calculated so as to make the resistance which the liquid offers plus the resistance of the springs such that the wheels will not jump from the ground when the piece is fired at zero elevation. This is accomplished by making the gravity moment of the system about an axis through the point of support of the trail greater than the sum of the moments of the piston pull and the spring resistance about the same axis. The recoil of the gun on the carriage is 48 inches. The counter recoil rod with one end secured to the cylinder has its free end centred and supported at all times in the bore of the piston rod. It regulates the velocity of return of the piece throughout the whole length of counter recoil, thus

obviating the sudden strain and shock in designs where the counter recoil is unrestrained until the piece is nearly in battery. As a result of this arrangement the gun can be fired without even as much motion as would dislodge a piece of money placed upon the wheel.

96. The shield is made of hardened steel .2 of an inch thick, in three parts. A small portion of the top is made to fold down so that it will not project above the wheels while travelling. With this arrangement the carriage might be overturned without injury. The shields are tested for acceptance by firing thirty caliber steel-jacketed bullets with muzzle velocity of 2,300 f. s. at a range of 100 yards. The bullet must not penetrate. The shield thus offers protection against small arm fire and shrapnel bullets.

97. The road brake has a double lever, one in front and one in rear of the shield. The first is for the use of the cannoneer seated on the axle seat as a road brake, and the other is placed where it can be readily operated in firing the gun to prevent movement of the carriage especially when fired from a sloping platform. The brake blocks bear upon the front of the wheel in the firing position where they are out of the way of the firing party. When limbered up in travelling on the road this places the blocks behind the wheels and no mud is collected on the blocks.

98. The sights comprise a panoramic telescopic sight on the left of the piece, and a range quadrant on the right. In using the range quadrant the gunner on the left may devote his entire attention to giving direction, while another cannoneer can give elevation at the range quadrant. The piece is, however, usually aimed by the gunner seated on the left, who has at hand the elevating and traversing wheels for training the gun. The sights are attached to the cradle, and since this does not move in firing the piece the sighting may be continuous. The panoramic sight enables the gun to be aimed by directing the sight upon any object off the line of fire, which is especially useful when any direct target made by the enemy is indistinct.

99. It is found in order to secure steadiness of the carriage that the time consumed in recoil and counter recoil must be made about two seconds. The operations of opening, loading and closing the block, and firing, consume about one second only, so that practically therefore the maximum rapidity of unaimed fire

from a piece of this description is about three seconds per round, or twenty rounds per minute. After the trail is set a rapidity of 10 to 12 aimed rounds per minute can be maintained for some time. As compared with this, the 3.2-inch gun would fire not more than about two rounds per minute.

100. The weight of the model 1902 carriage, exclusive of gun, is 1,308 pounds, as compared with 1,321 pounds for the carriage of the 3.2-inch rifle, which is remarkable in view of the additional parts in the new carriage, but shows how much the essential parts in the old carriage which are still retained in the new could be lightened and yet retain sufficient strength for service by introducing the plan of recoil of the gun upon the carriage.

101. Four rounds of ammunition are carried in tubes under the axle seats to be ready for immediate use, and 36 rounds are carried in the limber, Fig. 104. The gun and limber complete with ammunition weighs 3,800 pounds, giving a load of 633 pounds per horse for each of the six horses.

102. The organization of batteries with these guns will comprise four guns with carriages and limbers, and 12 caissons and limbers. The caissons carry 70 rounds each, and their limbers 36 rounds, making a total of 1,432 rounds for the battery, or 358 rounds for each gun.

Types of Guns, Mounts and Breech Mechanisms.

103. In all the more recent type of guns a continuous movement of breech mechanism in opening and closing the block is substituted for the separate operations, one to unscrew the block and the other to withdraw it, formerly required. The principal parts of the breech mechanism are the spindle with mushroom shaped head, plastic obturator and steel split rings to seal the escape of gas; the block with its threaded and smooth sectors arranged to require a small turning movement for locking; the parts that operate to turn and withdraw the block; the carrier (either tray or ring) that supports the block when open; the safety attachments that prevent firing until the block is closed and locked, and the firing mechanism.

104. Fig. 105 illustrates the more modern form of breech mechanism in the 6-inch rapid fire gun, model 1900. This gun gives 3,000 f. s. muzzle velocity with 100-pound projectile, is fitted with the Stockett breech mechanism and a firing mechanism

for combination electric-friction primer; and also a loading tray with automatic movement which rises to support the insertion of the charge and projectile when the breech is opened, and falls to clear the way for the block as the latter is closed. The capacity of the powder chamber in this gun is 2,114 cubic inches, as compared with 1,278 cubic inches in the model 1897 gun, which was designed for 2,600 f. s. muzzle velocity.

The 6-inch Bofors gun, Fig. 106, has a standard muzzle velocity of 2,624 f. s. with 100-pound projectile. The most interesting feature of the gun is the breech mechanism, which is opened and

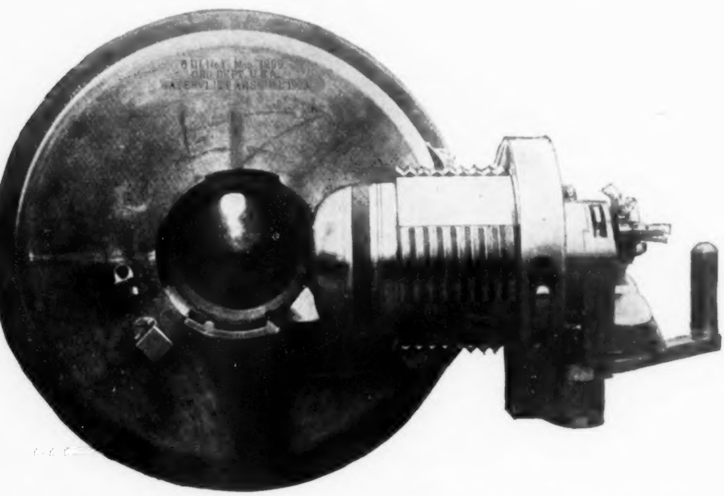


FIG. 105.—BREECH MECHANISM, 6-INCH R. F. GUN. MODEL OF 1900.

closed by a single movement and may be operated automatically or by hand. The projectile recess is conical in shape, with a large and convenient opening to the rear for loading, and is fitted with a loading tray that works automatically. The mounting is provided with two sets of sights, one on either side, so that one man can control the direction and another the sighting in elevation for rapid firing. In trials for rapidity at will 10 rounds were fired in 103 seconds with the breech operated by hand, and 10 rounds in 94.6 seconds with the breech operated automatically. The automatic arrangement is complete for both opening and closing the block. The opening is effected by the recoil of the gun in its cradle, which at the same time compresses a spring

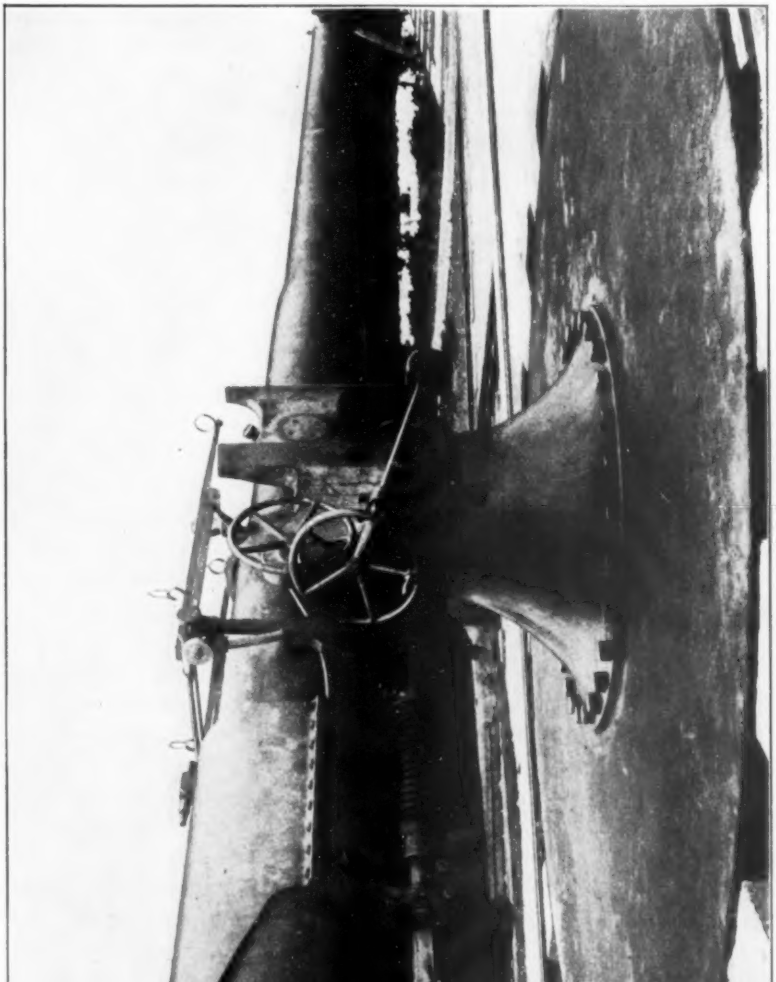


FIG. 106.—BOFORS 6-INCH R. F. GUN.

and a catch holds the block open. On releasing this catch the spring automatically closes the block. This mechanism has worked well throughout. The automatic feature gives only a little increase in rapidity of fire, but has the advantage of saving space for loading and dispenses with the services of one man. Among the many labor and time saving devices now attempted to be introduced for the service of guns, which are frequently objectionable because of delicacy and increased complications, the Bofors automatic breech opening and closing device appears exceptional for distinctive merit and certainty of operation. To see it in operation one experiences the same sense of relief, to a degree, as in the use of fixed ammunition instead of separate loading in the rapid fire field guns. The cannoneers are saved by so much from violent and rapid exertion, the loading progresses more smoothly and withal with increased rapidity.

105. This gun has been fired 386 rounds with a number of pressures exceeding 50,000 pounds, and in one case reaching nearly 70,000 pounds, derived from charges purposely increased to test the mechanism or the behavior of various lots of smokeless powder to determine if there existed a critical point of pressure.

106. Designs have been made, and work is in progress to apply plans of automatically opening and closing the breech block to other rapid fire guns and to guns mounted on disappearing carriages.

107. The 16-inch gun, Fig. 107, is at present mounted upon the proof carriage. It has been recently decided to make for this gun a Buffington-Crozier disappearing carriage, and to mount it at some place on the coast not yet designated. The piece has been fired eight times, using the first sample lot of DuPont nitrocellulose smokeless powder made for it, with charges varying in weight from 450 to 640 pounds and a standard weight of projectile of 2,400 pounds. Round No. 6 with 640-pound charge gave 2,345 f. s. muzzle velocity with 38,545 pounds pressure and muzzle energy 91,500 foot tons. Colonel Ingalls' estimate of the maximum range of this gun, provided it should be elevated for firing at an angle of nearly 42 degrees, is 20.9 miles, and the maximum ordinate of the trajectory 30,516 feet, or above $5\frac{3}{4}$ miles. The battle range of the gun with an elevation of 10 degrees, which may be given on the disappearing carriage, should be about 16,000 yards, or over 9 miles.

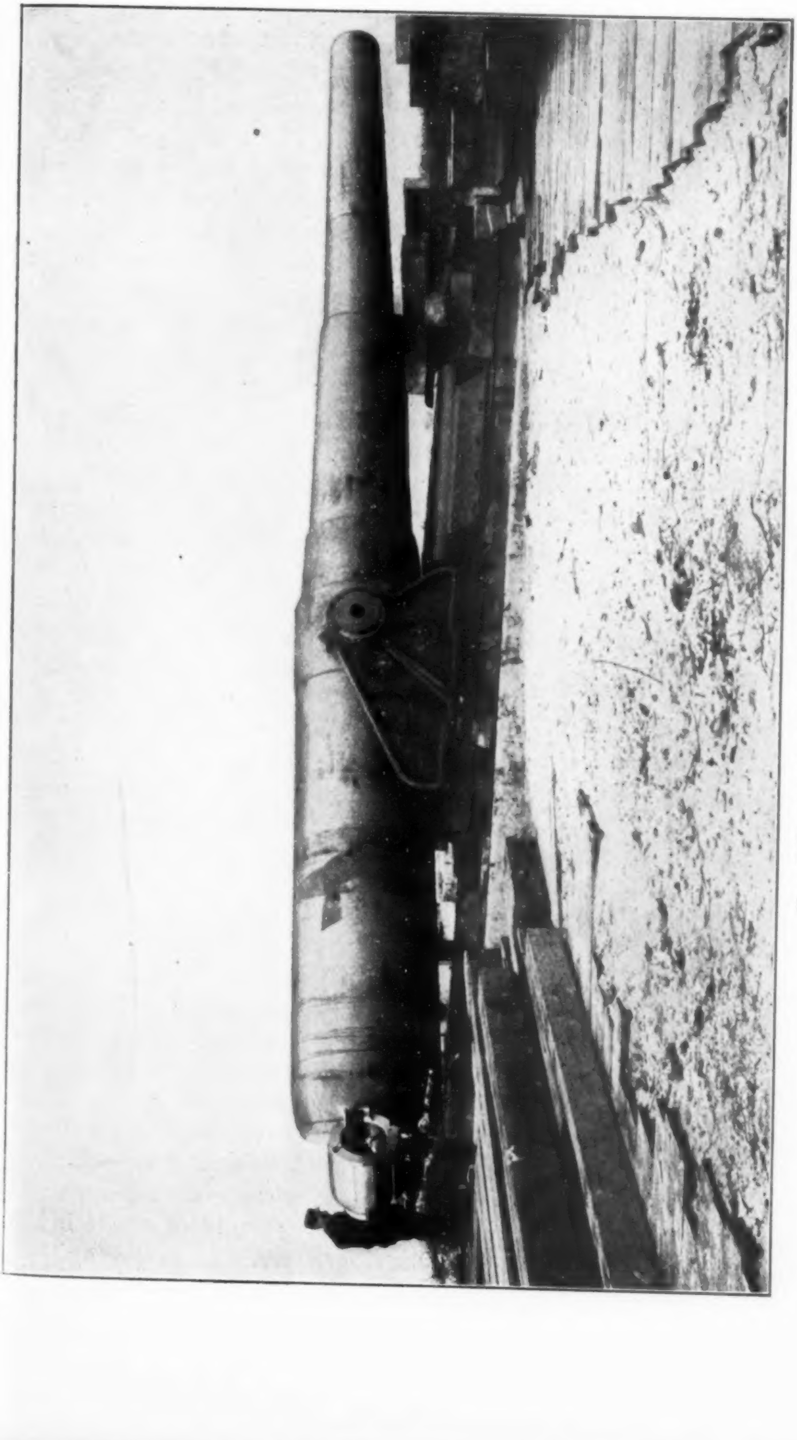


Fig. 107.—16-inch B. L. Rifle on Proof Carriage.

108. The proof firing was attended with entire success. At the fourth round, with a full charge, a velocity of 2,317 f. s. was obtained with 36,700 pounds pressure. The gun was designed and the powder sample made to give a velocity of 2,300 f. s. with not exceeding 38,000 pounds pressure. The results are remarkable when it is considered that the charge of smokeless powder exceeded largely any that had been heretofore fired from a gun.

109. The influence of caliber in producing power is shown in that the muzzle energy of the 16-inch gun is 2.4 times that of the 12-inch rifle, model 1895, designed for the same muzzle velocity. The 12-inch rifle, model 1900, with 2,560 f. s. muzzle velocity, gives a muzzle energy of 45,420 tons, or about one-half that of the 16-inch gun. In actual firings the 12-inch rifle, model 1900, has given a range of 13,360 yards, or 7½ miles with 10 degrees elevation.

110. Improvements in the appliances for giving elevation to 12-inch mortars comprise several designs of quadrants permanently fixed to the piece within view of the cannoneer at the elevating wheel, and also a range scale on a wheel attached to the elevating axle. The method formerly used, the quadrant placed on flats on the breech of piece, and requiring the gunner to give orders to the cannoneer at the elevating wheel to raise or lower the piece so as to adjust the level of the quadrant was slow and cumbersome. With the new arrangements the elevation is given by bringing the pointer to a fixed point on an arc, and does not depend upon the adjustment of the level bubble. Fig. 108 shows the 12-inch mortar carriage, model 1896, with improved quadrant in place over the right rimbase.

Powders.

111. Smokeless powder only is now manufactured for the service of all guns, and the black or sphere-hexagonal and brown prismatic powders are discontinued except to use up material on hand for target practice. However, a quantity of fine grain black powder is used as a priming charge in the smokeless powder cartridge. The present practice makes this priming charge 2.6 of one per cent. of the smokeless powder charge, and is the cause of producing a considerable amount of smoke in the discharge, particularly with the large guns.

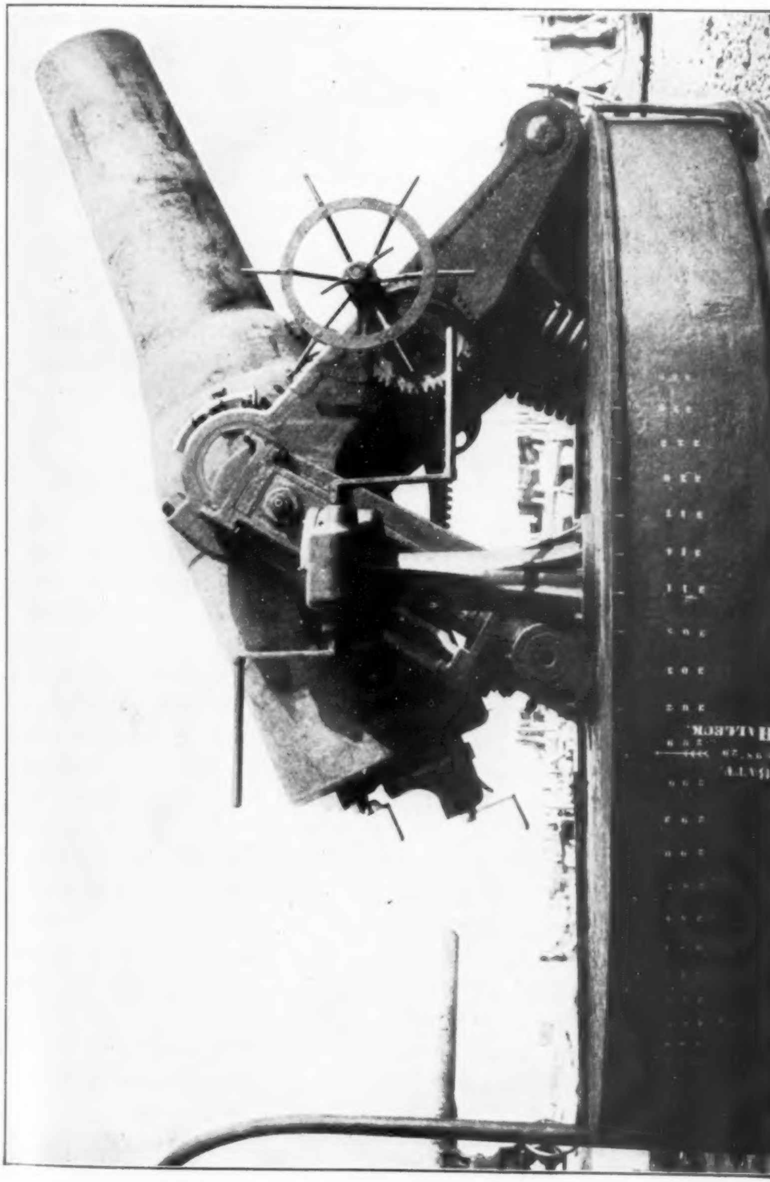


FIG. 108.—12-INCH B. L. MORTAR AND CARRIAGE.

112. The form of grain now in use is a multi-perforated cylinder which when burned in the open air gives an increasing surface of combustion until the circles meet in burning away, leaving 12 to 16 per cent. of the grain in splinters to be consumed with a decreasing surface of combustion. Other forms in use elsewhere are the strip and the hollow cylinder or tubular grain. The latter burns away with a uniform surface of combustion, and this, combined with its facility for ease of loading when made up into cartridges, appears to make it the most desirable form to use.

113. Nitrocellulose powder has generally displaced that containing nitroglycerine. The heat produced is less and larger charges are required, but this is offset by the decreased erosion found with the nitrocellulose composition.

114. The advantage of smokeless over the brown powder, aside from the question of smoke, lies in the fact that it is all converted into gas, while the brown and black powders give somewhat more than one-half the products of combustion in solid residue. All the properties of smokeless powder are not as yet known, and there is a large field for investigation. Recent experiments have shown that certain lots of powder at least have a critical point of pressure. That is to say, the pressures will be found to increase regularly according to a certain law with the charge up to a certain point, beyond which, if the charge is increased, very high and abnormal pressures may be encountered. These experiments have been purposely made with charges greater than the service requirements in order to investigate the law of critical pressures.

115. Smokeless powder requires a large air spacing, and the powder chambers of guns to give high velocities in order to hold the large charges required for this must be made correspondingly large. There is one size of powder grain best adapted to each caliber of gun, and nothing is gained by increasing the size or in effect using a slower burning powder for that gun. A larger charge of the slower burning powder would be required to give the same velocity, with loss of efficiency. For an equal weight of charge the gas will have a longer path to work over and will impart more velocity to the projectile the nearer to the breech the powder is consumed in the bore. The limit for minimum size of grain and quickness of burning is fixed by the pressure that the gun will support in the powder chamber, and the limit

for maximum size of grain may roughly be evidenced by the absence of unconsumed powder on firing.

Projectiles.

116. The present approved form of shrapnel body comprises a steel case with bursting charge in the base, and the case made strong enough to eject the balls after the manner of a small gun without rupture when the bursting charge is ignited. The base charge gives an increase of about 200 f. s. velocity to the balls over that of the shrapnel at the point of burst. The cone of dispersion is about 9 or 10 degrees. In the shrapnel made with the bursting charge in the head the cone of dispersion is somewhat greater. The velocity of the balls, however, is somewhat retarded by the burst and the effective range is materially less than that of the shrapnel with bursting charge in the base.

117. The common form of armor-piercing shot and shell have capacity to perforate about one caliber of hard-faced steel armor for the shot, and one-half caliber for the shell. Both are to be used in our service charged with high explosive. The 12-inch shot, for example, holds 23 pounds and the 12-inch shell 68 pounds of explosive. They are made of special quality of steel containing chromium or manganese, and carefully hardened and tempered. The processes of manufacture are held as trade secrets by the different makers.*

118. Where armor penetration is not required, as in high explosive shell for the field and siege guns, the walls of the shell are made as thin as consistent with necessary strength to withstand the crushing effort accompanying discharge from the gun so that a maximum charge of explosive may be carried.

119. Recently an armor-piercing projectile, with thickness of wall intermediate between those of the present shot and shell, has been produced by the Wheeler Sterling Company of Pittsburg. This projectile, by virtue of the improvements in manufacture, has the same armor-piercing power as the thick-

---

* Figs. 109 and 110 show the characteristic bursting effect of the two high explosive compounds that have been adopted in our service. The projectiles exploded were 12-inch armor piercing shell, which weigh, empty, about 950 pounds, and contain about 67 pounds of Maximite, or 57 pounds of explosive *D*, with Frankford Arsenal detonating fuze. The shell were buried ten feet in sand for explosion, and the fragments afterwards recovered as shown.

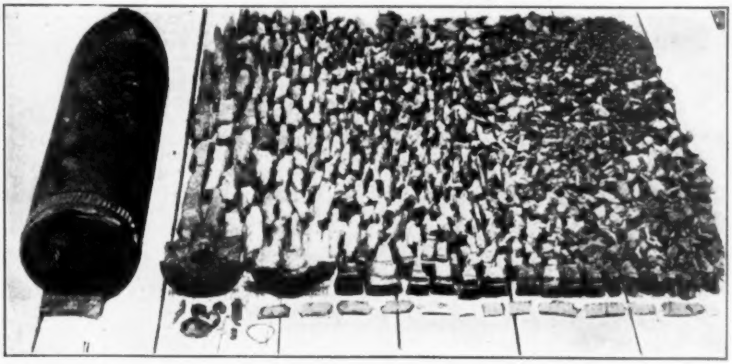


FIG. 109.—FRAGMENTATION, 12-INCH A. P. SHELL CHARGED WITH MAXIMITE.

walled shot and costs about the same as the shot but nearly twice as much as the present shell. It is designed to replace the two present projectiles and give a single armor-piercing projectile of good bursting charge capacity for each gun. The base of the projectile is cut away to allow the band to be stripped at the rear in perforating armor and thus diminish the resistance to penetration due ordinarily to the enlargement of the diameter of the projectile at the band. The cap is assembled by means of a wire forced into a groove made partly in the projectile and partly in the cap after the latter is in place. The cap of the present projectile is forced and hammered into a groove on the projectile.

120. A striking example of the effect of low temperature on steel plates was seen in firings a few days since at the Proving

FIG. 110.—FRAGMENTATION OF 12-INCH A. P. SHELL CHARGED WITH EXPLCSIVE D.

Ground. 8-inch A. P. shell were fired at a tempered, nickel steel plate 9 feet x $7\frac{1}{2}$ feet x 3 inches thick, backed by oak timbers. The weather had remained at a temperature approaching zero for several days. The projectiles were fired to strike the plate with angles varying from 20 to 40 degrees between the face of the plate and the line of fire, to observe the effect at these small angles on the plate and also to test the relative efficiency of capped and uncapped projectiles.. At each round the cold plate was badly cracked, and markedly more so than a similar plate previously tested in warmer weather. Thereupon two pieces from the same plate were heated to a temperature of 100 degrees or more and attacked by similar projectiles. In both cases, although the plates (pieces) were smaller than before, each piece was dished and bent by the impact of the shell but there was no cracking. These tests have so far negatived the report received from English sources that the uncapped projectile is equal in proficiency to the capped projectile at very oblique angles of impact. On the contrary it appears that the capped projectile is decidedly superior.

Accuracy and Endurance of Guns.

121. It is the practice of the Ordnance Department to subject each gun to an exhaustive test for endurance before adopting it as a type for construction and introduction into service. All the heavy sea-coast guns are hooped to the muzzle.

122. The estimated tangential resistance of the 8, 10 and 12 inch guns, that is, the pressure per square inch which can be supported in the powder chamber without exceeding the elastic limit of the gun, is about 52,000 pounds, and that at the muzzle about 22,000 pounds.

123. If these pressures are exceeded a permanent set of the bore may be produced, but rupture is prevented by the ductility of the metal and the guns can support higher pressures. The standard limit of powder pressure for the charge is 38,000 pounds per square inch, or about 73 per cent. of the elastic resistance of the gun.

124. The actual limit of safe pressure for these guns is probably about 70,000 pounds per square inch. But one case of explosive rupture of a gun of this type has occurred at the Sandy Hook Proving Ground aside from experiments with explosives. This was a 10-inch gun in March, 1899, using an experimental

smokeless powder which proved to be of brittle nature. The breech of the gun was destroyed. The two pressure gauges registered 78,000 and 79,000 pounds per square inch.

125. On another occasion the breech block of a 5-inch gun was stripped under an estimated pressure of 70,000 pounds. 6-inch R. F. gun No. 1, model 1897, sustained a pressure of 86,000 pounds as registered by the gauge, which resulted in wedging the breech block. Various calibers of the guns, including the 12-inch, have been repeatedly subjected to pressures of 50,000 pounds, and in exceptional cases considerably higher pressures without serious effect.

126. The type guns of heavy caliber have sustained the following number of rounds for endurance, namely:

Experimental	8-inch gun No. 1, West Point foundry........	335 rounds
"	8-inch gun No. 1, Watervliet Arsenal type....	390 rounds
"	10-inch gun No. 1, Watervliet Arsenal type...	292 rounds
"	12-inch gun No. 1, Watervliet Arsenal type...	265 rounds

and the types of 12-inch mortars, models 1886 and 1890, respectively, 398 and 399 rounds. Barring erosion, all the guns are intact except the second 8-inch, in which longitudinal cracks appeared in the tube near the muzzle after 388 rounds.

127. Plans have been prepared for relining the 10-inch gun, No. 1, by a half tube inserted from the breech, which will remove the eroded part of the bore and should render the gun fit for a duplication probably of its first record.

128. The following accuracy targets, see also Fig. 111, form part of the record of the guns named:

Gun.	Rounds numbered.	Range.	Deviation from centre of impact. Mean horizontal.	Mean vertical.	Mean.	Rectangle Containing shots.
		Yds.	*Ft.*	*Ft.*	*Ft.*	*Ft.*
10-inch B. L. R., W. A., No. 1.....	(8) 97 to 104	1,760	1.94	0.88	1.28	3.4 × 4
" " " "	(5) 283 to 287	3,000	1.44	2.61	2.98	8.2 × 4.4
10-inch B. L. R. Crozier wire wound	(5) 271 to 275	1,760	1.84	0.64	1.95	3. × 7 2
12-inch B. L. R., W. A., No. 1....	(5) 216 to 220	3,000	1.64	1.68	2.35	6 8 × 5.2

129. It will be observed that three of the targets were made, beginning with round 216 in the 12-inch gun, round 271 in the

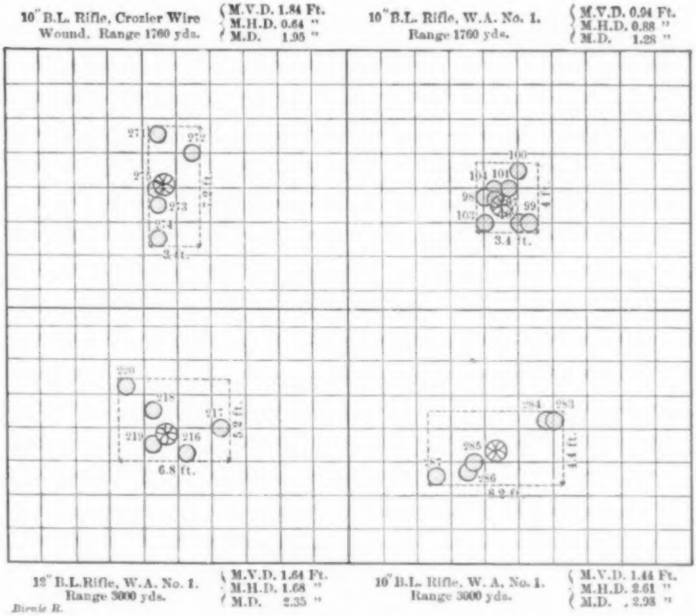


FIG. 111.—ACCURACY TARGETS, 10- AND 12-INCH B. L. RIFLES.

10-inch Crozier wire wound gun and round 283 in the 10-inch forged steel gun. Before these firings the guns had become so eroded that the original bands on the projectiles were too small to seat the projectile or take the rifling properly. New and enlarged bands were used in each case with the result of restoring the accuracy of the guns practically to the original standard.

The Disappearing Carriage.

130. The utility of the disappearing carriage as a system was seriously attacked in 1899 by the Major General commanding the army, followed by the Board of Ordnance and Fortification over which he presided at that time. On October 8, 1900, the Board, by a majority vote of the members present at the time, recommended that no additional disappearing carriages be manufactured for use on high or medium sites, and that no more should be manufactured for use on low sites until the proportion of those to be placed on such sites should have been reduced to one-third of the

total number to be so located. The objections were directed generally: first to the use of disappearing guns, and second to criticisms of the then adopted type of disappearing carriage. The objections and criticisms on both points have been very fully and completely answered by papers published in the reports of the Chief of Ordnance, U. S. Army, for 1900, appendix 32, 1901, appendix 58 and 1902, appendix 1. These references can be consulted by any desiring to acquaint themselves with all the bearings of the subject. The question was finally settled (appendix 1, Report of 1902) by the report of a board of seven members appointed by the President pursuant to act of Congress. The membership of this board comprised Colonel Wallace F. Randolph, Chief of Artillery; Captain Eugene H. C. Leutze, U. S. Navy; Major John G. D. Knight, Engineers Corps; Major Charles Shaler, Ordnance Department; Major Albert S. Cummins, Artillery Corps; Captain William H. Coffin, Artillery Corps; Mr. John R. Freeman, of Providence, R. I., with Captain Richmond P. Davis, Artillery Corps, as recorder. The board was directed to test the system by firing not only the thirty rounds from a 10-inch disappearing gun prescribed by Congress, but also thirty rounds from a 10-inch barbette and ten rounds each from 6, 8 and 10 inch guns mounted on both disappearing and barbette carriages. The board visited five artillery posts and fired over 150 rounds with full service charges, and in conclusion recorded its opinion that the general mechanical principles involved in the chief elements and movements of the Buffington-Crozier disappearing carriage are admirably adapted to their purpose. In these trials 30 continuous rounds were fired from the 10-inch disappearing carriage in 27 minutes 10 seconds. The average interval between rounds being 54.3 seconds, the shortest interval 47 seconds.

The more this question is studied in connection with the increased accuracy and power of gun fire from shipboard the more reason there appears for affording the best possible protection for the gun and its mount, and the gun crew as well. This requires guns to be mounted either on disappearing carriages or else to be thoroughly protected by shields. One gun that can be kept in action at a critical time is evidently better than a number that are so exposed as to be silenced or seriously damaged in action. The straight front shield usually employed on barbette carriages affords a very limited amount of cover, and a

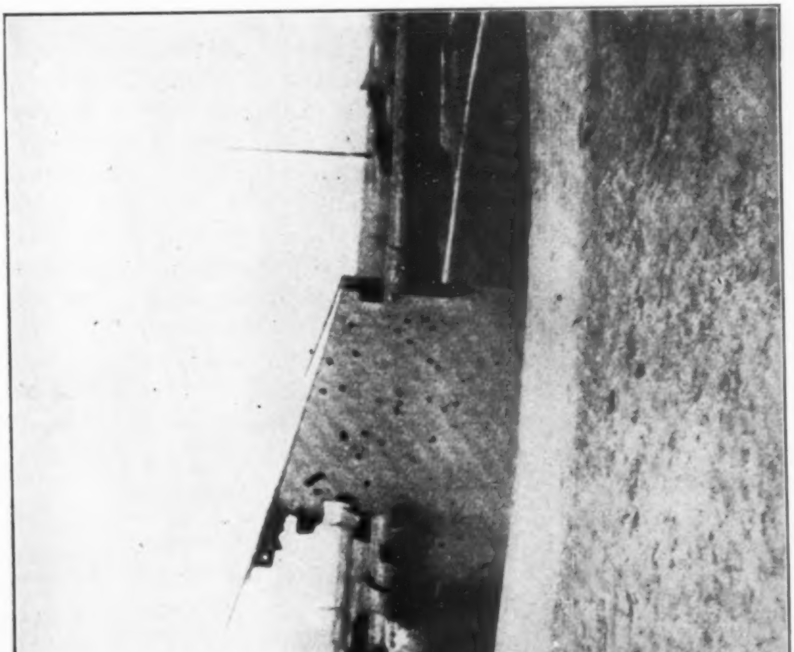


Fig. 112.—Barbette Mount, Taku Forts, China. After Battle.

shield thin enough to be penetrated by shell is a source of danger rather than protection.

131. The illustration, Fig. 112, showing an Armstrong gun in the Takau forts, China, in 1900, is well in point. They were attacked by gunboats carrying rapid fire guns only at ranges between 2,000 and 3,000 yards. The inner sides of the shields were covered with blood when examined after the capture of the forts, and around one gun were found the bodies of 35 dead Chinamen. One gun was put out of action by a projectile that entered under the chase and injured the mechanism of the carriage. This is the same pattern of shield regarding which the testing board of 1902 remarked, "With the non-disappearing Armstrong carriage with shield the gunner has better protection than he has on any other mount examined by the board." The 4.7-inch Armstrong gun shields have a thickness of 4 inches at the front and one or two inches on the flanks. The 6-inch Armstrong carriage shield is 4½ inches thick at the front. The quality of the steel is such, however, that even the fronts can be perforated by 5 and 6 inch rapid fire guns at battle ranges.

132. The thickness of shields prescribed for our barbette carriages is 1½-inch for 6-pounder, 2 inches for 15-pounder and 4½ inches for all larger calibers, including the 5 and 6 inch rapid fire and the sea-coast barbette carriages. This steel is required to be face-hardened of the best quality. The curved shield recently tested on a dummy 6-inch barbette pedestal mount, model 1900, is made of face-hardened, Harveyized steel 4½ inches thick. It was able to keep out projectiles of 5-inch and smaller calibers, but was perforated by a 6-inch projectile fired with a reduced velocity to simulate a battle range of about 3,000 yards. The cost of this shield, about $10,000, scarcely compensates for the partial protection it affords.*

133. In firing from a disappearing carriage the sighting of a gun in direction and elevation is given before the gun rises into the firing position, when it can be immediately fired and the gun remains exposed to hostile fire for a few seconds only.

* Fig. 113 shows the 6-inch barbette carriage shield with the axis of the gun turned 60 degrees from the line of observation and after 9 rounds had been fired against it, namely, one 3.2-inch; five 5-inch; and three 6-inch projectiles. The perforation seen in the side exposed to view was made by a 6-inch capped A.P. shot fired with a striking velocity corresponding to about 3,100 yards range and at an angle of 50.5 degrees between the line of fire and a tangent plane to the plate at the point of impact.

FIG. 113.—45-INCH SHIELD, BARBETTE MOUNT, 6-INCH GUN AFTER PROOF-FIRING.

134. A very important attribute of the disappearing carriage besides the protection of the gun mount and gun crew from hostile fire, is the concealment it affords on the approach of an enemy as compared with a barbette carriage and shield. When the vicinity of the disappearing battery is masked by trees or shrubbery, which should be planted if the natural features are bare, the disappearing battery becomes most difficult to identify at a distance, while a barbette gun is always in evidence and when furnished with a shield becomes very conspicuous.

135. The model 1901 12-inch disappearing carriage, Fig. 114, is controlled by electric motors in manœuvring, and combines other improvements. The time for raising into battery is reduced to about six seconds, or less than one-half the time required with the previous models of 12-inch carriage. The traversing and elevating are controlled by the gunner on the sighting platform through electric motors, and the telescopic sight is so supported in a bracket geared with the elevating apparatus of the gun that the gunner can himself give the necessary elevation for range with the same facility as with guns on pedestal mounts. The telescopic sight has a 3-inch triple objective, 15-inch focal length, Brashear-Hastings erecting prisms and two eye pieces giving powers of 12 and 20 with a field of 4 and 3 degrees respectively. The eye end of the telescope will be provided with a rack and pinion for focusing and with adjustable cross wires. Small electric lights are provided for night use to illuminate the scale diaphragm in the telescope and the deflection and elevation scales of the instrument. The sight includes a range drum graduated in yards. These sights and the telescope are furnished by the firm of Warner & Swasey, Cleveland, Ohio.

Methods of Range Finding and Fire Control.

136. The methods of fire direction in the sea-coast service include three cases. One, where the gunner aims directly at the target without predicted range and direction, and continues to fire as rapidly as possible by observing as far as practicable the errors of previous shots. This method is used generally with rapid fire guns. Two, where the gunner gives direction to the gun by observing the target through the telescopic sight, and the gun is elevated to give a predicted range obtained from the range-finding instruments. Three, when the range and direction

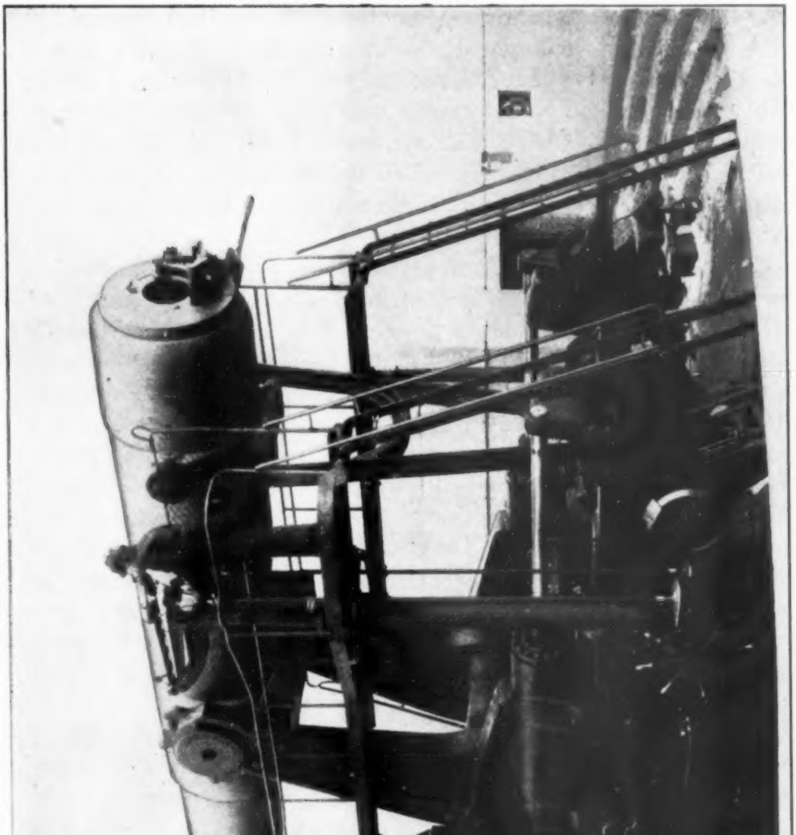


FIG. 114.—12-INCH BUFFINGTON—CROZIER DISAPPEARING CARRIAGE. MODEL OF 1901.

are both obtained by the range-finding appliances and transmitted to the gun, which is aimed accordingly. In this case it is not necessary for the gunner to see the target. The case applies to all mortar fire and to the special case of salvo points, the range and direction of which are known beforehand, and the guns can be pointed to be discharged when the moving target reaches a salvo point. Cases two and three are of most general application, especially for distant fire.

137. The ranges are first obtained by either the depression or horizontal base range-finding instruments, and by means of a plotting board and accessories in the battery commander's station are reduced to the position of the gun that is to be fired and then transmitted by telephone and telautograph to the battery officer at the gun. When the gun is reported ready for firing the battery commander discharges it by pressing an electric button at his station connected by wire with the firing mechanism and electric primer in the gun. Should the electric firing fail a cannoneer stands ready to pull a lanyard and fire the primer by friction. In case of a moving target the predicted range and direction sent to the gun are calculated to set the gun sufficiently ahead of the target so that the projectile will reach the target when it arrives at the predicted point. The time of flight of the projectile is necessarily involved in this prediction.

138. This is a very brief outline of the methods employed which involve in practice a number of instruments and accessories, and a very complete system of governing rules for fire direction and for fire control by the commander of a group of batteries.

139. Two methods of measuring ranges are employed. One, by the depression range finder and one by a horizontal base with azimuth instruments at each end. The depression range finder being complete in itself affords much greater convenience than the horizontal base in operation. The difficulties with the horizontal base are especially great in recognizing the target; that is, to insure that both observers are looking at the same vessel. The limitations in the accuracy of the depression instrument, however, which must be mounted at a height of at least 60 or better 100 feet above sea level, coupled with the difficulty of securing this elevation at many sea-coast forts—unless by building expensive and very conspicuous towers—necessarily lead to the use of the horizontal system in many cases. The system now being

worked out is based upon a judicious combination of the two classes of instruments.

140. The Warner & Swasey depression range finder, Fig. 115, has been constructed with full knowledge of the requirements of an instrument of this kind and fulfills these requirements better than any other known instrument for the purpose. It has been tested at the Proving Ground as well as by artillery officers at various posts in target practice and is now being supplied in quantity for the service. In effect, when the instrument in any given station has been adjusted for height of tide and for refraction it enables the distance to a target to be read directly from the yard scale by bringing the horizontal cross-wire on the waterline of the object. The same firm has also designed a self-contained horizontal base measuring instrument. A model of one-fourth size with a base six feet three inches long has been tested with satisfactory results.

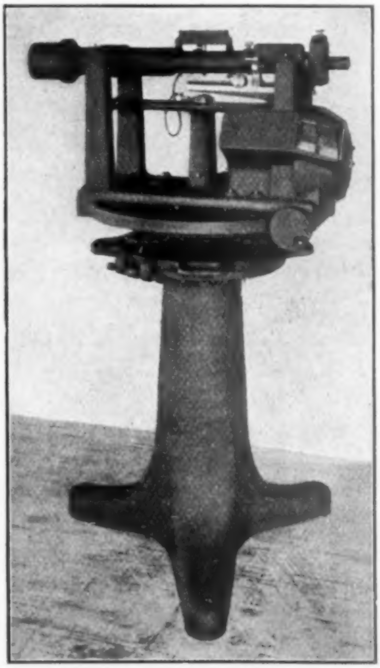


FIG. 115.—WARNER & SWASEY DEPRESSION RANGE FINDER

PAPERS

OF THE

CHICAGO MEETING

(XLIXth)

OF THE

AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

MAY 31st TO JUNE 3rd, 1904.

No. 1028.

PROCEEDINGS

OF THE

CHICAGO MEETING

(XLIXth)

OF THE

AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

May 31st to June 3rd, 1904.

LOCAL EXECUTIVE COMMITTEE.

ROBERT W. HUNT, *Chairman.*
LOUIS MOHR, *Chairman Finance Committee.*
GEORGE M. BRILL, *Chairman Entertainment and Excursion Committee.*
WM. L. ABBOTT, *Chairman Hotel and Reception Committee.*
FRANCIS W. LANE, *Chairman Printing Committee.*
W. H. V. ROSING, *Chairman Transportation Committee.*
PAUL M. CHAMBERLAIN, *Chairman Meetings Committee.*
J. H. WARDER, *Secretary Local Committee.*

The forty-ninth meeting of the American Society of Mechanical Engineers was made noteworthy by the fact that it was a joint meeting of the American Society with the Institution of Mechanical Engineers of Great Britain. The Council of the American Society had invited the British Institution, by a formal vote, suggesting that the latter body make their spring meeting of 1904 an American Meeting, with a view of visiting the great exposition in St. Louis which would be in progress in commemoration of the centennial of the purchase of Louisiana from the French in 1803.

This invitation was cordially accepted by the Council of the Institution of Mechanical Engineers and arrangements were at once begun by the executive officers of the two institutions to carry out the common purpose of such a joint meeting.

It was recognized that a very significant feature of the visit of the British Engineers to the United States would be the opportunity and privilege of visiting the American installations and study American practice in the line of their particular interests.

To this end the American Society had organized headquarters in the various cities to which the visiting engineers were directed by accrediting letters, and through which channels the arrangements for the local visits were provided. The visiting engineers had come to America sufficiently far in advance of the date of the meeting to accomplish their purposes in part before the meeting, and were arranging to visit other cities after the meeting.

The date for the joint meeting was set for May 31st to June 3rd, and in the city of Chicago, rather than in St. Louis, in order that the advantages might be reaped from the ample and convenient hotel accommodations in Chicago, and so that the real business of the meeting might be completed before the distractions and interests of the exposition should compete for available time of the visitors.

The headquarters of the Society were opened on the morning of May 31st, in a room in the Auditorium Hotel, and it became at once obvious that the numbers in attendance from American and British sources would be unusually large. The total registration during the four days exceeded the highest figures reached in any previous gathering, and were approximately as follows:

American Society	350
British "	75
Ladies and guests	518
Total	943

The registration system was carried out by a card principle in multiple, and this meeting was the first at which a satisfactory method was used for the carriage by each member of his name as a feature of the lapel badge of distinction of member and guest.

The name was used only by members of the two societies and a different color for the card was used for the American and British members. The plan worked smoothly and acceptably.

The opening day was given over to incidental excursions to the underground subways of the city and to miscellaneous and individual interests.

The first session was arranged to be held in the Music Hall of the Auditorium for Tuesday evening.

Opening Session. May 31st, 8.30 p. m.

The first session was held in the Music Hall of what is designated as the Fine Arts Building, and was called to order by Mr. Robert W. Hunt, of Chicago, Chairman of the Local Committee. On the platform with Mr. Hunt were Mr. Ambrose Swasey, president of the Society; Mr. J. Hartley Wicksteed, president of the English Institution; Mr. Edgar Worthington, secretary of the English Institution, and Prof. F. R. Hutton, secretary of the American Society.

Mr. Hunt introduced the Comptroller of Chicago, the Hon. Lawrence E. McGann, who had been delegated by the corporation to represent it in welcoming the engineers as guests of the city.

Mr. McGann spoke in detail of the obligation which the city recognized to the work of the engineer, and referred to the financial obstacle which had prevented progress in these directions at the rate which the city would have been glad to follow.

President Swasey of the American Society spoke of the Society meeting once again in Chicago, referring to the meetings in 1886 and in 1893, and commenting on the growth of the Society during these intervals He spoke of Chicago being the only city where the Society has ever held three of its mid-year meetings so far. He made his welcome at this point include particularly the pleasure of the American Society in welcoming the British Institution to a joint session under such favorable circumstances.

President J. Hartley Wicksteed of the Institution of Mechanical Engineers of Great Britain expressed his pleasure in taking part in the welcome of the engineers, and gave reference to the pleasure of the visitors in coming to the various cities of the United States, and noting the differences which were characteristic of them.

At the close of the responses of the respective presidents, a letter was read from Mr. James Forrest, past-secretary of the Institution of Civil Engineers of Great Britain, in which he expressed the wish that the joint meeting might strengthen the bonds of international fellowship and social union between the two great branches of the English-speaking race.

At the close of the formal exercises the meeting adjourned to an informal reception, tendered by the local residents, in the parlors of the hotel, at which a light collation was served, and which gave opportunity for the members to renew old acquaintances

and form new ones, and for the members of the two societies to come into social contact.

SECOND SESSION. WEDNESDAY, JUNE 1ST, 10 O'CLOCK A. M.

The second session was called to order in the large ball-room of the Auditorium Hotel on the ninth floor, overlooking the lake. The meeting was called to order by President Swasey of the American Society, who took occasion to report to the meeting the action of the Council under the provisions of the Constitution, on the previous afternoon, whereby, after nomination in due form, Past-President John E Sweet, one of the founders of the Society, had been elected to honorary membership.

He spoke in fitting words of Professor Sweet as a capable man, an able engineer, and that if any living member of the Society was entitled to the name of "Father of the Society of Mechanical Engineers" it was the member whom the Society had honored, and in honoring whom it honored itself. The announcement of this election was received with applause.

The Secretary reported for record the Report of the Tellers of Election as follows:

REPORT OF TELLERS OF ELECTION.

The undersigned were appointed a committee of the Council to act as tellers under By-Laws 6, 7, and 8, to scrutinize and count the ballots cast for and against the candidates proposed for membership in their several grades in the American Society of Mechanical Engineers, and seeking election before the Forty-ninth Meeting, Chicago, 1904.

They have met upon the designated days in the office of the Society, and have proceeded to the discharge of their duty. They would certify for formal insertion in the records of the Society, to the election of the following persons, whose names appear on the appended list in their several grades.

There were 661 votes cast on the ballot ending May 14th, 1904, of which 50 were thrown out on account of informalities. There were 596 votes cast on the ballot ending May 21, 1904, of which 32 were thrown out because of informalities. The tellers have considered a ballot as informal which was not endorsed, or where the endorsement was made by a facsimile or other stamp.

CHARLES E. LUCKE,
HENRI G. CHATAIN, } *Tellers of Election.*

May 14, 1904.

To be Voted for as Members.

Allen, W. M.
Andrew, J. D.
Angus, J. J.
Arnold, George
Baker, J. C. W.
Bausch, F. E.
Bidle, W. S.
Coster, M.
Darlington, P. J.
Davol, G. K.
Diemer, H.
Dunbar, W. O.
Gerhard, W. P.
Goddard, D.
Harris, W. A.
Hood, O. P.
Howells, C.
Jenness, C. H.
Kelman, J. H.
Kenyon, A. L.
Lafore, J. A.
Larkin, W. H., Jr.
Lenfest, B. A.
Lord, J. E.
MacMurray, J. T.
Meier, K.
Metcalf, L.
Muckle, J. S.
Powell, J. E.
Randall, D. T.
Safford, A. T.
Sage, H. M.
Sanborn, F. E.
Smith, A. L.
Smith, P. F., Jr.
Stillman, G. F.
Teele, F. W.
Toltz, M. E. R.
Trampe, J. A. C. L. de
Walker, G. S.
Wells, E. C.
Wells, J. H.
Wolf, F. W.

For Promotion to Full Membership.

Bump, B. N.
Hannah, F. A.
Logan, J. W.
Marks, L. S.
Wallace, D. A.

To be Voted for as Associates.

Gillette, J. W.
Johnson, W.
Judd, Horace
Logan, J.
Ludeman, E. H.
Merrill, J. L.
Norton, F. L.
Noyes, H. T., Jr.
O'Neil, J. G.
Parkhurst, F. A.
Rippey, S. H.
Williams, G. W.
Woods, F. F.

For Promotion to Associates.

Libbey, J. H.
Robbins, C. C.

To be Voted for as Juniors.

Buckingham, H. H.
Canby, H. B.
Diederichs, H.
Dudley, S. W.
Gamon, E.
Gillett, W. L.
Glasgow, C. L.
Jackson, P.
Kelley, G. C.
Kenney, L. H.
King, R. S.
Knowlton, F. K.
Lockett, K.
Massie, J. H.
Matthews, F. E.
Morgan, L.
Murphy, J. K.
Nate, E. H.
Newbury, G. K.
Norris, H. L.
Powell, E. B.
Rantenstrauch, W.
Richards, C. D
Schaeffer, S.
Schlemmer, E.
Scott, E. F.
Slawson, H. H.
Snodgrass, J. McB.
Sponsler, C. F.
Stacy, H. W.
Townsend, H. P.
Wachalofsky, C. J.
Woodward, R. S., Jr.
Young, C. H.

MAY 21ST, 1904.

TO BE VOTED FOR AS MEMBERS.

Bennett, C. W.	Jones, B. N.	Rivett, Edward.
Clemens, A. B.	Loomis, B., Jr.	Schulte, G. H.
Corbett, R. H.	Lucas, H. M.	Slade, A. J.
Elmes, C. W.	Massa, R. F.	Smith, E. J.
Haven, H. M.	Monteagle, R. C.	Stivers, W. D.
Holdsworth, F. D.	Nicholson, S. T.	Taylor, C. L.
	Reid, Joseph.	

FOR PROMOTION TO FULL MEMBERSHIP.

Belsley, Clay.	Doty, Paul.	Price, A. M.
Bouton, G. I.	Hutchison, R.	Watts, Geo. W.
Collier, W. H.	Jacobs, Ward S.	Whitaker, H. E
	King, J. H.	

TO BE VOTED FOR AS ASSOCIATES.

Beebe, M. C.	Nelson, J. W.	Pond, H. O.
Garred, U. A.	Obert, C. W.	Stevens, H. L.
Hill, H. H.		Van Winkle, E.

FOR PROMOTION TO ASSOCIATE MEMBERSHIP.

Langoltz, Robt.	Morrison, H. S.

TO BE VOTED FOR AS JUNIORS.

Barnard, E. E.	Heaton, H. C.	Knoop, T. M.
Chasteney, C. D.	Hodge, Geo. O.	Midgley, F. W.
Cobleigh, H. R.	Horton, W. H.	Sibley, M. M.
Gardner, Henry		Sibson, H. E.

Under the provisions of the Constitution and By-Laws, the Chairman announced these gentlemen in the foregoing list to be duly elected to membership in their respective grades.

Mr. Wilfred Lewis, Chairman of the Commitee on Standard Forms of Machine Screws, reported in a letter that very satisfactory progress had been made, and the committee confidently expected to complete its work in ample time for the presentation of a report at the annual meeting of the Society in December.

The Chair then called upon the Committee representing the Society in the planning and labor concerning the proposed engineering building which Mr. Andrew Carnegie had projected for the use of the societies who should take part in its advantages. This report, in the absence of other representatives of the Society,

was presented by the Secretary, and is made a special appendix to the Proceedings of this meeting.

At this point the President called for new business, motions and resolutions.

Mr. Fred J. Miller, of New York, presented at this point a proposed amendment to the method of amending the Constitution of the Society. The debate on this amendment was as follows:

Mr. Fred J. Miller.—In accordance with paragraph 59 of the Constitution, I wish to offer an amendment to that paragraph. Perhaps I had better read the paragraph as it is in the Year Book, so that if any discussion takes place we may discuss it intelligently. May I do that?

President Swasey.—As you please.

Mr. Fred J. Miller (reading).—" At any semi-annual meeting of the Society any member may propose in writing an amendment to this Constitution. Such proposed amendment shall not be voted on at that meeting, but shall be open to discussion and to such modification as may be accepted by the proposer. The proposed amendment shall be mailed in printed form by the Secretary to each member of the Society entitled to vote, at least sixty days previous to the next annual meeting, accompanied by comment by the Council, if it so elects. At that annual meeting such proposed amendment shall be presented for discussion and final amendment, and shall subsequently be submitted to all members entitled to vote, provided that twenty votes are cast in favor of such submission." That means twenty votes at that meeting. " The final vote on adoption shall be by sealed letter-ballot, closing at twelve o'clock noon on the first Monday of March following."

Paragraph 58 reads:

" The letter-ballot, accompanied by the text of the proposed amendment, shall be mailed by the Secretary to each member of the Society entitled to vote at least thirty days previous to the closure of the voting. The ballots shall be voted, canvassed and announced as provided in the By-Laws. The adoption of the amendment shall be decided by a majority of the votes cast. An amendment shall take effect on the announcement of its adoption by the presiding officer of the semi-annual meeting next following the closure of the vote."

I call attention to the fact that according to this paragraph of the Constitution if a member propose any amendment to the Constitution he need not secure the approval of any other member

of the Society. It does not even require that the proposed amendment shall be seconded. If it is merely proposed at a semi-annual meeting, in writing, though it may then be discussed, a vote upon it is prohibited at that time; it may only be discussed and the proposer may accept amendments if he choose, but no matter if he is unable to secure the approval of a single other member of the Society for the proposed amendment, the Secretary is, nevertheless, compelled to print it, at the expense of the Society, and to send it to every member of the Society to be voted upon. Then it must come back to the Secretary, must be again discussed, and if it then receive twenty votes in the meeting it must be again sent out to every member to be voted upon a second time, and, if it then receives their approval, it is adopted. Now, in case no amendment is made at the second presentation of the amendment, that is, if no modification of it is made, the Society is placed in the position of sending out to the full membership identically the same thing to be voted upon a second time. That is to say, though it has been sent out once and approved, it must be again sent out for a second vote, even though no alteration whatever may have been made.

In place of those two paragraphs I propose the following:

AMENDMENT TO THE CONSTITUTION.

"At any semi-annual meeting of the Society any member may propose in writing an amendment to this Constitution. Such proposed amendment shall then be open to discussion and to such modification as may be accepted by the proposer, and if it then receives in its favor 20 votes from among the members present it shall, at least 60 days previous to the next annual meeting, be mailed in printed form by the Secretary to each member of the Society entitled to vote, accompanied by commments by the council if it so elects. The vote upon the amendment shall be by sealed letter ballot closing at 10 A.M. on the first day of the annual meeting first succeeding the semi-annual meeting at which the amendment was proposed. The ballots shall be voted, canvassed and announced as provided in the by-laws, and if a majority of the votes cast are in favor of the amendment it shall, after declaration of that fact by the presiding officer thereupon take effect and be in force."

It seems to me that this amendment obviates the difficulty into which this clause of the Constitution as it now stands may lead us. I have, I may say, some other amendments which I think should be made, but I do not present them now, because I do not wish to propose them under this law as it now stands, as I would be compelled to propose them without their having received the approval of any other member, and I wish simply to get this under

way in order that neither myself nor any other member who wishes to propose an amendment may be compelled to propose it in the manner in which the Constitution now requires it to be done.

President Swasey.—Gentlemen, you have heard the amendment proposed by Mr. Miller, which is now open for discussion. What have you to say in regard to it?

Mr. R. H. Soule.—Having been a member of the Committee on the Revision of the Constitution I am familiar with the circumstances and considerations which led up to the adoption of the particular clause which it is now sought to amend; there was not one article of the entire Constitution and By-laws for which so many alternative arrangements were offered as this one; the committee spent a great deal of time in discussing the best method of amending the Constitution, and finally decided on the form which has since been adopted; and the underlying principles in reaching that conclusion were these:

First of all it was considered wise to give every member of this Society a right to stand up on this floor and to suggest an amendment to the Constitution, and it was purposely made to read so that it was not necessary for him even to secure a second; it was considered that it ought to be the fundamental right of every man to submit an amendment to the Constitution, and at least claim a hearing for it; the other idea was that it would be a sound principle that the main debate on that proposed amendment should be precipitated into the arena of discussion at the annual meeting of the Society; the fact is recognized that the annual meeting is always the largest, and therefore it is the one at which it may be assumed that an amendment would receive the most careful consideration and at the hands of the largest proportion of the membership.

That is the logic which pervaded the minds of the members of the Committee on Revision of the Constitution, and which led up to their suggestion of that particular clause which it is now proposed to amend.

Mr. Fred J. Miller.—In regard to what Mr. Soule says as to the right of every member to propose an amendment, I would be the last one to deny that right, for I most thoroughly believe in it; but I call attention to the fact that my amendment toes not really abridge the right of a member to propose an amendment; he still has the right to propose, but there is no use in putting the Society to the expense of printing his amendment and sending it out to

the membership if he cannot get twenty members to agree with him that the amendment is a desirable one; because it is then morally certain that his amendment cannot be adopted, and the bother and expense of the general vote will be to no purpose.

I have studied the Swiss Initiative and Referendum to some extent. Any citizen of the Swiss Republic has a right to propose a law, but it is there clearly seen that it would be folly to arrange so that any Swiss citizen could simply write out a proposed law, and that it should then be immediately submitted to every member of the Government to be voted upon. The Swiss Constitution requires that the proposer of the law shall have back of it the support of a certain number of citizens first secured by the proposer. That is what we want here. Another point which occurs to me is this: That an amendment may be sent out to the membership to be voted upon, and it may be approved by them; the members expressing their desire to have the Constitution so amended, and yet that amendment comes up in the next annual meeting of the Society, and is discussed if at that time it fails to receive twenty votes, the will of the Society is set aside, *i.e.*, although the membership at large has voted in favor of the amendment it cannot be adopted, simply because it cannot be sent out for a second vote by the membership at large. I do not think that is right.

President Swasey.—Is there any further discussion? If not, the amendment will take the regular course laid down by the Constitution.

No other new business being presented the meeting took up the professional papers as follows:

The paper by Mr. Harrington Emerson on "A Rational Basis for Wages" was discussed by Messrs. Emerson Bainbridge, E. J. Chambers, and J. Hartley Wicksteed of the British Institution, and Mr. H. H. Suplee of the American Society.

Two papers presented by British authors were then read by the Secretary of the British Institution.

The paper by Mr. George Watson was entitled "The Burning of Town Refuse," and the paper by Mr. C. Newton Russell was entitled "Refuse Destruction by Burning, and the Utilization of Heat Generated." President Wicksteed opened the discussion, which was participated in by Mr. Alfred Saxon, Mr. Charles Wicksteed, and Mr. G. R. Dunell of the British Institution.

At the close of these papers the meeting took a recess until the evening of the same day.

Third Session. Wednesday Evening, June 1st, 8.30 p. m.

The third session was called to order in the Music Hall of the Fine Arts Building by President Ambrose Swasey. The papers of this evening were specifically devoted to the Power Plant Problem and Practice, especially on the development of the steam-turbine. In the discussion of the three papers, Messrs. Storm Bull, George I. Rockwood, D. S. Jacobus, H. H. Suplee, George W. Colles, H. L. Doherty, and Alex. Dow took part. Messrs. Hodgkinson, Rearick, Rice, and Kerr spoke in the closures. The full titles of the papers were as follows: "Some Theoretical and Practical Considerations in Steam-Turbine Work," by F. Hodgkinson; "The De Laval Steam-Turbine," by E. S. Lea and E. Meden; "The Curtis Steam-Turbine," by W. L. R. Emmet; "Different Application of Steam-Turbines," by A. Rateau; "The Potential Efficiency of Prime Movers," by C. V. Kerr.

Fourth Session. Thursday Morning, June 2nd.

The session was opened by the presentation of two papers on the Tall Office Building Problem, by Messrs. Bolton and J. H. Wells. These papers were respectively entitled "The Power Plant of the Tall Office Building," and were discussed by Messrs. Bryan, Colles, Rockwood, Suplee, Bunnell, Gifford and Mr. Nistle of the American Society.

The next paper was from the English Society, and was entitled "The Middlesbrough Dock Electric and Hydraulic Power Plant," discussed by Messrs. Barr, Alfred Saxon, and John Etherington; the paper entitled "Use of Superheated Steam and of Reheaters in Compound Engines of Large Size," by Mr. Lionel S. Marks, was discussed by Messrs. Barrus, Kerr, and Rockwood. The paper by Mr. William P. Flint, entitled "Commercial Gas Engine Testing and Proposed Standard of Comparison," was discussed by Messrs. Mathot and Chambers of the English Society, and Professor Jacobus of the American.

At the close of the discussion the meeting took a recess until the final session on Friday morning.

Final Session. Friday Morning, June 3rd.

The Joint Meeting was the guest for this session of the Lewis Institute of Chicago, West Madison and Robey Streets. The session was introduced by the presentation by Secretary Worthington, on behalf of Mr. William Campbell, of the report of the

Institution's committee on "Effects of Strain and Annealing," prepared by Dr. William Campbell. This report was presented in abstract by the use of lantern slides projected upon a screen.

Following the presentation of this report, which was commented on by President Wicksteed, the paper by Mr. William J. Keep was read on "Cast-Iron, Composition, Strength and Specifications," which was discussed by Messrs. West and Emerson of the American Society and Mr. Chambers and Mr. Crosta of the British Institution.

At this point, President Wicksteed took the chair for the presentation of the paper by Prof. J. T. Nicholson on "Experiments with a Lathe-Tool Dynamometer," which was presented by Mr. Daniel Adamson of Manchester, England, with whom he had been associated. In the discussion, Messrs. Wicksteed, Emerson, McGeorge, Benjamin, and Pilton took part.

President Swasey of the American Society resumed the chair for the presentation of the papers on locomotive testing, by Messrs. Hitchcock and Goss, respectively, entitled: "Locomotive Testing Plants," by Prof. W. F. M. Goss; "Road Tests of Consolidation Freight Locomotives," by Prof. E. A. Hitchcock.

Messrs. Bement and Worthington took part in the discussion.

With this group of papers the presentation of technical matter was completed.

Director Carmen of the Lewis Institute introduced at this point the Hon. C. C. Kohlsaat, Judge of the United States Court in Chicago and president of the Board of Trustees of the Lewis Institute. Judge Kohlsaat expressed the pleasure of the Trustees of the Institute in having the body its guests for the morning, and referred in brief to the desire of the Institute to make itself useful in the spread of engineering education and in fitting young men to carry on the work of the world in industrial lines.

Mr. Robert W. Hunt spoke in feeling words of the death during the progress of the convention of Mr. David R. Fraser, a veteran member of the Society resident in Chicago, who had been a member of the Local Committee of Arrangement, and who was a foremost representative of the industrial interests of the city. Due memorial action was to be taken by the Society in its published volume of Transactions.

President Swasey at this point read from the chair the following minute:

The American Society of Mechanical Engineers, speaking for its own members, desires at the outset to record its pleasure in the fortunate outcome of its plans

which have brought the Institution of Mechanical Engineers from England to be its guests and co-workers in the pleasures and labors of its Chicago Convention. It was a happy thought of Mr. James Rowan of Glasgow to present to the Americans the possibility of holding a meeting of the Institution in the United States. It is with great pleasure that the Americans have been permitted to receive so many members of the Institution and to take them into all the elements, both professional and social, of such a successful convention. The Americans hope that their guests will carry home with them a still more earnest recognition of the fact that the two branches spring from a common stock by race and language, and that the laws and practice of engineering must necessarily be one on both sides of the Atlantic.

At the close, President Wicksteed of the British Institution rose and took the chair, and presented the following minute on behalf of the British Society:

The Institution of Mechanical Engineers, as represented by its President, Council, visiting members and Secretary, desires to take occasion on this last session of the joint meeting to pass a resolution which shall convey to the American Society the warm thanks and recognition which they desire to express for all that has been done for their pleasure and profit during the American visit of the Institution. They desire to thank the Council of the American Society and the executive officers who represent it for courtesies, official and personal, and for the arrangements which have been made for them, the visiting engineers, to have opportunities of studying American industry in its home field.

The Institution desires further to record the pleasure its members have derived from the attentions and opportunities which have been extended to them during their stay in Chicago, and in their joint participation in the Chicago Meeting. The visitors ask that, in the resolutions of thanks for specific things which are about to be presented to the joint meeting, they may be particularly included, and that they as well as the society may be identified with these resolutions.

At the close of this action, the Secretary of the American Society presented on behalf of both organization the following resolutions of thanks and recognition, which were unanimously adopted:

1. *Resolved*, That the American Society of Mechanical Engineers and the Institution of Mechanical Engineers desire to express to the City of Chicago and its city Government the sincere thanks of the joint meeting for their courtesy, extended by a representative of the City in the person of its Comptroller, in addressing a welcome to the visitors at their opening session on Tuesday evening. They would express their pleasure in having brought to their notice the recognition which the Corporations was kind enough to express of the debt which Chicago owes to the engineering talent which has helped to make the City what it is.

2. The American Society and the Institution of Mechanical Engineers have received from the Illinois Steel Company at South Chicago a notable courtesy in the permission and arrangements for the joint visit through the Illinois Steel

Works on Wednesday afternoon. The Mechanical Engineers recognize what is meant by a permission of this sort and the difficulties in handling so large a number through the complicated and dangerous departments of a productive establishment of this magnitude. They ask that the Illinois Steel Company and Mr. W. A. Field, the General Superintendent, and other officers of the Company will accept the sincere thanks of the party for the success attending the visit, and for the admirable way in which the visit and entertainment were planned.

3. The joint session of the American Society of Mechanical Engineers and the Institution of Mechanical Engineers have receievd from the Illinois Central Railway most notable courtesies, entertainment and provision during their Chicago visit. They would ask that that Company and in particular Mr. W. H. V. Rosing, Assistant Superintendent of machinery, will accept the sincere thanks of the Engineers for their tender of free transportation over their lines during the Chicago Convention, and for the special and admirable arrangements whereby the Company and its agents have served the convenience of the visitors on their excursions and visits about the City.

4. The joint meeting of the Mechanical Engineers of England and America ask that Marshall Field and Company will accept the sincere thanks of the societies for the opportunity given to the ladies and members to visit the wonderful establishment which is known by their name, and for the courteous arrangement for the luncheon of the visitors on the afternoon of Thursday at that great mercantile establishment. The character of the hosts on this occasion is a factor in economic and commercial development which all engineers are watching with the keenest interest in its bearing upon the problem of economic handling between producer and consumer.

5. The joint meeting in Chicago has been made memorable for the Mechanical Engineers by the particular importance which has been given at this meeting to the subject of the Steam Turbine. The Engineers, therefore, ask that the Commonwealth Electric Company and the Chicago Edison Company will accept the sincere thanks of the visitors for the privilege of the visit to the Fisk Street Station and the chance to see the working of a power station of this magnitude. They would couple this vote of thanks in particular with the name of Mr. Louis A. Ferguson, Second Vice-President of the Company, and with those of Mr. Abbott and members of the local committee who have been instrumental in bringing about this pleasant result.

6. Those members of the joint meeting who have taken occasion to visit the Stock Yards and the Packing House of Swift and Company, ask that that Company and Mr. Edward S. Swift in particular and other officers of their corporation will accept sincere thanks for the invitation to visit their great undertaking, and to study their admirable, humane and careful methods which have been followed in the prosecution of their business and in the attainment of their wonderful results.

7. To the Trustees of the Art Institute and to Mr. H. N. Carpenter, Secretary, the Engineers beg to extend their most sincere appreciation for the distinguished courtesy accorded by these gentlemen, in permitting the reception of Thursday evening to be held in the midst of the attractive collections of that Institute. The relation between engineering and art is a close one, even if by many its existence is not detected. Art is a child of economic progress and wealth. Engineering is the foundation of real productive wealth. It has given the Engineers great pleasure that their reception should be accorded to them in the midst of such congenial and appropriate surroundings.

8. The Engineers ask that the Trustees and Faculty of the Lewis Institute of Chicago will accept the sincere thanks of the joint meeting for the invitation to hold its closing session in the Hall of the Lewis Institute. The Engineers recognize what the profession owes to the work of the technical educators and are glad to record by this resolution their hearty sympathy in the work of the Institute and their recognition of the far-reaching effect of such education, both in industry, in civilization and in culture. They would express in particular their thanks to Professor P. M. Chamberlain for his instrumentality in bringing about this pleasant result, and heartily appreciate the courtesy for the entertainment and luncheon which the Trustees have provided.

9. It is to the Trustees of the Sanitary District that the Mechanical Engineers owe the courtesy of an invitation to inspect the Drainage Canal, a great work in the interests of Chicago, which has been prosecuted under their direction. The thanks of the Societies and the Institution are also due to the Atchison, Topeka and Santa Fe Railroad Company for the courtesy of the special train provided by that Company through Mr. Kendrick, third Vice-President of the Railway system. The visiting Engineers ask also that the Western Society of Engineers, by whose active co-operation this excursion has been originated, will accept the sincere appreciation of the visitors for their share in the successful visit.

10. Beginning with the very first day of the attendance of visitors and continuing beyond the official end of the meeting have been the courtesies of the Illinois Tunnel Company. The Engineers desire to return thanks to the Company and to Mr. George W. Jackson in particular for the courteous invitation to visit the tunnels and to study under such favorable circumstances their possibilities and their working. The Mechanical Engineer is entirely wonted to the experience that some of his most successful achievements are where the eye of the superficial observer knows nothing of their real excellence. That we should have had the opportunity to go under ground and under competent guidance study the hidden detail is an experience for which we desire to record our thanks.

11. It must be known to any Mechanical Engineer that the success which comes to an acceptable product of a machine depends upon the skill, the forethought and the careful administrative attention of the brain of the Engineer that has originated the machine. In the successful meeting of a Convention, such as our Chicago meeting has proved itself to be, the Engineer behind the product is the local committee of resident members in whose hands we have been but raw material, and which have succeeded in turning out so successful a finished product as the Chicago meeting. The visiting Engineers ask that Mr. Robert W. Hunt, with a membership common to both societies, who has acted as chairman of the local committee, will accept for himself and for his associates a sincere recognition for his and their share in bringing this delightful result to pass. They ask also that Mr. George M. Brill, efficient and capable Chairman of the Entertainment and Excursion Committee, who has abandoned his home that he might be near his responsibilities, will allow himself to be included in our warm expression of thanks and recognition. The meeting would not have been the success which we have enjoyed, had less gifted, interested or assiduous hands been at the throttle.

12. While the ladies are important and appreciated factors as captains of industry in the control of the individuals who pass resolutions, it is unfortunate that they do not have a speaking voice in the proceedings of the official part of this meeting, and that from "seconds in command" and not from the captains

themselves must come the voicing of the thanks which the ladies of our party must tender to the efficient and delightful group of ladies organized under the local committee for the pleasure of the visitors. This resolution is to give voice to the official thanks of the visiting ladies for the drive and the luncheon on Wednesday, for the organization and direction of the charming visit on Thursday and for the crowning delight of the drive on Friday. The ladies ask also that Mrs. Robert W. Hunt will accept this unsatisfactory effort of mere men to express for them the thanks of the ladies for the delightful reception on Thursday afternoon.

The men and the women leave for home with delightful memories of new friendships formed, of old friendships strengthened and a memory of a week full of pleasure, satisfaction and profit.

These resolutions being put by the chair they were unanimously adopted.

At the close of the sessions the Society and its guests were entertained by the ladies of Lewis Institute at a most satisfactory luncheon on the top floor in the department of the Institute devoted to domestic science, and afterwards took a train on the Atchison, Topeka and Santa Fe Railroad for the controlling works of the drainage canal.

On the way the party visited the distributing yards of the railway companies. The party were escorted by float a short distance on the canal itself, and after visiting the work of the controlling outflow, re-embarked on the train and returned to Chicago.

In the evening, the Society and its guests were again entertained by the Local Committee, the entertainment taking the most attractive form of an orchestral concert in the Auditorium Theatre by the Thomas Orchestra of Chicago.

This concert was made noteworthy by its ending. A precentor came forward on the stage, and led the audience in two verses each of the American national hymn, "My Country 'Tis of Thee," and the English form of the same air, "God Save the King."

With this incident the meeting officially closed.

The next meeting of the Society will be its regular annual meeting, to be expected in the city of New York the first week in December.

No. 1029.

APPENDIX.

CARNEGIE GIFT TO ENGINEERING.

THIRD CIRCULAR.

In former circulars the members of the Society have been advised that at the Saratoga Meeting in June, 1903, there was reported in official form the proposed gift to engineering by Mr. Andrew Carnegie, a member of the Society, and that at the New York Meeting in December, 1903, a supplementary report was made of the work of the Society's Committee up to the beginning of that year. The first report will be found on page 870 of Volume XXIV., as paper No. 976, and the second on page 34 of Volume XXV., as No. 1008.

At the Chicago Meeting in May, 1904, a further report of progress was presented for the information of the members as a feature of the business session of that meeting. The report was presented by the Secretary of the Society, who is one of the representatives of the Society on the Committee, and was as follows:

It will be recalled by those who have followed the history of the undertaking that in February, 1903, Mr. Andrew Carnegie made his first offer of a million dollars in order that the four national engineering societies might be housed in a proper building, which should give necessary accommodation for the executive offices of the engineering societies, and should contain auditoriums for meetings and give adequate space for the libraries of the societies, the plan also to include accommodations under a separate roof, but as a part of the general scheme, for the Engineers' Club, which is the social organization having no necessarily professional outlook.

The action which was taken on the receipt of Mr. Carnegie's proposition was to appoint a Conference Committee, consisting of three members from each of the four national societies and from the Engineers' Club. This Conference Committee held

frequent meetings for the consideration of its problem, and discharged its duties under a provisional organization until March, 1904, when by the letter-ballot of the American Society of Civil Engineers—for reasons satisfactory to themselves—it was decided by that body that they would not partake in the benefit of Mr. Carnegie's gift.

The Conference Committee was then at once faced with the problem: Would the donor give his proposed gift to the three engineering societies and the Engineers' Club, and would the three societies be able to make themselves responsible for the maintenance and conduct of the building, or would the problem become greater than these three bodies could cope with?

The matter was immediately laid before the Councils of the three societies and on careful consideration of the resources of each body, the three organizations, the American Society of Mechanical Engineers, the American Institute of Electrical Engineers, and the American Institute of Mining Engineers, decided that it would be within their power to resume this responsibility.

Careful estimates of the cost of operating the building were prepared, and by special meetings the decision was reached that each society was prepared to go forward in co-operation with the other two. When this action was communicated to Mr. Carnegie, with characteristic promptness of decision, he at once wrote another deed of gift, stating his willingness to present to the three societies and the Engineers' Club a building which should cost a million and a half dollars, in recognition of the larger scope of the proposition as submitted to him by the three societies.

It is the purpose of the Trustees that this building should provide for all organizations which have engineering as their basis in the matter of suitable auditoriums, executive offices, and most of all, for their libraries.

Mr. Carnegie's letter of gift has, therefore, constituted the representatives of the three engineering societies and the Engineers' Club, an organized committee of twelve, who for the moment are to administer what has been called the "Carnegie Trust for Engineering." This committee of twelve has the responsibility for the expenditure required for adequate buildings. While the gift is as one, there are in reality two buildings. The three societies are to occupy a frontage of 125 square feet on the north side of Thirty-ninth Street in New York City, with a depth of 100 feet. The Engineers' Club is to occupy a site 50 feet wide by 100 feet on the south side of the Fortieth Street frontal of the same city

block. The buildings do not quite come to each other in the relation of the upright and the arm of a capital letter " T," because the Engineers' Club stands nearer one end than the other of the long frontage of the Engineering Building.

The Conference Committee, in its capacity as Trustees, at once made arrangements to have the design of the building competed for and the architects selected by the competitive method.

A programme of competition was prepared with some elaborateness, deciding a number of questions in advance in order that the competition might be on the common basis. Among these questions were, first, that the auditorium which is a central feature of such a proposed engineering building should have such a capacity that the best voice of an author or speaker could be distinctly heard in every part of it. This brought to the front at once an interesting question as to the floor space and height within which an untrained speaker taking part in the meetings could be heard and understood. It was plainly absurd to make the auditorium so large that discussions were a mere dumb show of moving lip and gesture. It seemed on conference with theatrical people, ministers, architects, and public speakers, that such ordinary speaker could be heard if the audience did not exceed one thousand persons on the floor, with a possible additional five hundred seated in a commodious gallery.

The committee would be very glad to receive additional light on this question from any members who may have experience to offer.

There are also to be smaller auditoriums having a capacity of two hundred and fifty persons and downward, so that smaller meetings shall not be lost in a large auditorium, and so that meetings of different societies of different sizes can be held at the same time. Such smaller auditoriums will also be cosy for monthly reunions.

The auditorium must, necessarily, by ordinance of the Building Department of New York City and as a matter of common sense, be located near the level of the street.

Above the auditoriums will be the office floors, one for each of the societies, giving accommodation for its reception rooms, offices for secretary, accountants, stenographers, and similar administrative departments, and the meeting room for councils and committees.

At the most liberal calculations for the present needs of the societies, a floor area of 5,000 square feet would seem to be ample.

Each floor will furnish over one-and-a-half times this space, so as to leave room for growth and future needs. It has been quite interesting to note that the three societies, calculating independently, ask the committee for practically the same amount of floor space.

Above the three floors allotted to the founders' societies can be certain additional floors, if the funds available shall make this possible, which shall be assigned to the use of the other engineering organizations which will naturally be the beneficiaries of Mr. Carnegie's gift. Such bodies are the Electro-Chemical Society, American Gas Light Association, Electric Light Association, and others. These accommodations will not be rented, but will be furnished to the users on a basis of a pro rata of operating expenses. It is plainly unfitting that there should be any profit-making in the conduct of the enterprise from a commercial point of view.

On the top floor or floors, which will probably be about the twelfth floor from the street, will be located the consolidated libraries and the reference and reading rooms.

It is an interesting fact that when the three societies met together by appointed meetings to consider the library problem in an engineering building of this character, it was at once the sense of these representatives of the three societies in their conferences, that there was before them the opportunity of a lifetime to create a great engineering library, with the strength and interest and the knowledge and personnel of the three societies behind it. The representatives of the New York Public Library, which will be located in a building just across the street, were called in at this conference, and their advice asked on the possibilities of a scheme of federation, whereby the New York Public Library and the Engineering Library could be mutually serviceable.

By locating the reading-room on the top floor it will not only be more quiet, but it will be in the better air, above the fly-line, and it will obtain light from the roof.

The Committee feel emphatically that this should be a working library, in which workers and students should be able to have access to the shelves and the books themselves.

To secure this greater development of the library idea will make a demand upon the societies themselves, but it is the belief of the Committee that the societies will be glad to enlist increasing energy and effort in return for what the library development will bring.

If the plans of the Committee are realized the library of this building will be The Engineering Library of the Atlantic Seaboard.

In carrying out the plans of the competition, the Committee have secured Professor William R. Ware as Professional Adviser on architectural questions. He is perhaps of all American architects one who has had the widest experience in judging competitions of the character which the Committee has had in mind.

With the advice of Professor Ware, and what could be gotten from other sources, the printed Programme of Competition in pamphlet form was issued on the 5th of May.

In view of advice given us from many sources it seemed plain that six weeks was a long enough time to allow architects for the consideration of their preliminary plans. In this view, counting forward from the 5th of May, the competition was announced closed on June 20th.

The competition is of the character which is known technically among architects as "a mixed competition," in that six firms have been invited to compete, and will be compensated, whether successful or not by a fee of $1,000. Besides this the competition is open to any architect in good professional standing, who has been in practice under his own name for over two years, to send in competitive drawings, and he will receive consideration exactly the same as the salaried or invited firms. All are compelled to compete under an absolute incognito. Four prizes of $400 each will be the award to the four most successful plans in the "open class."

Of course, the competition means nothing for the present but the selection of the architect, who will prepare the plans finally approved by the Committee. It does not mean in any sense that the plan of the successful architect is to be the plan approved by the Committee in final form. The Committee reserves the right to change and modify and remake the drawings of the successful competitor until its own ideals shall be realized.

It may be of interest to state that seventy-five applications have been received, and while we know that some of these writers do not intend to compete, but simply ask to recieve the competition because it was a cleverly prepared document, yet the Committee feels sure that there will be a sufficient number of competitors to make an interesting variety when the Committee comes to decide. Nothing can be done in the way of actual construction until after the first of July of the current year, since the site of both build-

ings is now occupied by residences, the leases on which will only expire on that day.

The buildings must be removed and the work of final settling on details of plans must follow before the actual construction can be begun.

The Committee has also had under consideration the problem of conducting the building after it should be completed and the construction problems have been solved. The present Committee are the Building Committee, and their functions will end with the completion of the building.

This has been met by appointing a Committee on Organization, through whom a charter has been secured from the State of New York, and signed by the Governor, May 11th, 1904.

The general scheme of organization provides for the election of trustees by the three societies, who enter the undertaking as founders, under the deed of gift.

The Engineers' Club will manage its building independently as soon as construction is completed.

These trustees are to be elected by the Councils of the individual societies, and will be three in number from each body, the term of one expiring each year. These trustees will have the responsibility of financing and managing the administrative detail of the building as a whole for the benefit of the founders' societies themselves, and the other engineering societies who will be participants in the privileges of the building.

The by-laws of the body provided by the charter will have to be very carefully prepared with advice of counsel, to provide for many questions in advance which are incident to the creation of a trust of this character.

The Committee proposes to proceed slowly with the consideration of these questions of organization, but hope that before the annual meeting they will be provided with a full and carefully thought-out scheme which will be reported to the Society at that time.

F. R. Hutton, *Secretary.*

No. 1030.*

THE USE OF SUPERHEATED STEAM AND OF REHEATERS IN COMPOUND ENGINES OF LARGE SIZE.

BY LIONEL S. MARKS, CAMBRIDGE, MASS.

(Associate Member of the Society.)

1. The object of this paper is to collect and present to the Society the results of a number of unpublished tests made during the past five years on several high-speed, two-cylinder compound engines, all built by the same makers, and all of the same type. The engines tested differ from one another only in size, in cylinder proportions, and in their working conditions. The investigations were made to determine the performance of the engines under different loads, both with and without jacketing and reheating. A comparison of the results for the different tests throws some light upon the influence on the thermal efficiency of large sized four-valve compound engines of the following factors:

(*a*) The use of a reheater.

(*b*) The use of moderately superheated admission steam.

(*c*) The load.

(*d*) The size of the engine.

(*e*) The cylinder proportions.

2. The results recorded here are for tests made on nine separate engines and for twenty-eight different tests. The engines are parts of three electric lighting plants situated in or near Boston.

Engine *A* is at the L Street station, South Boston, of the Edison Electric Illuminating Company—a plant which at the time of the test was the property of the Boston Electric Light Company. The tests at this station were conducted by the writer.

Engines *B*, *C*, *D*, *E* & *F* are at the Atlantic Avenue station of the Edison Electric Illuminating Co., and they were tested

---

* Presented at the Chicago meeting, May and June, 1904, of the American Society of Mechanical Engineers, and forming part of Volume XXV. of the *Transactions*.

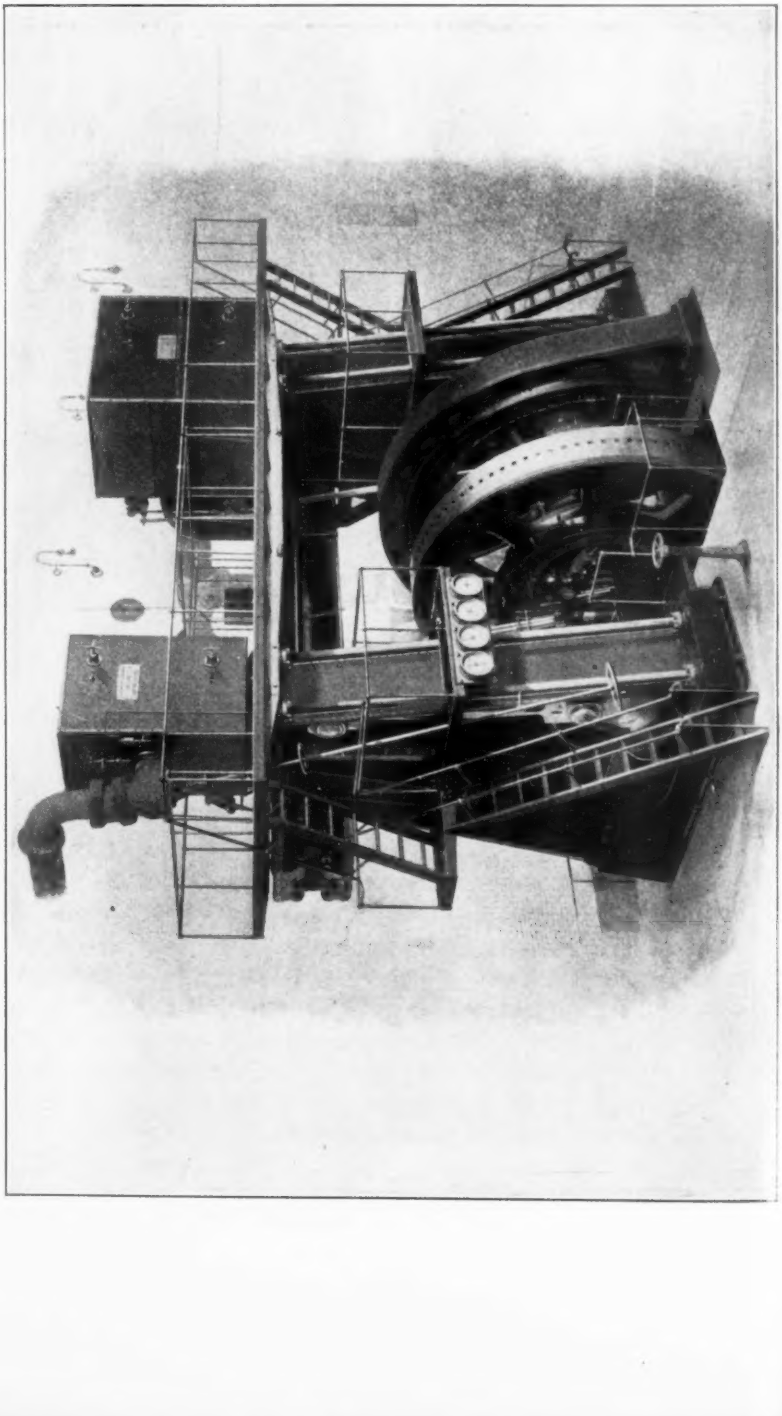


FIG. 116.—ENGINE A.

under the joint supervision of an employe of the company and a representative of the engine builders. Messrs. C. H. Parker and H. Cook, both members of this Society, were in charge of most of the tests, but in some tests their places were taken by Messrs. C. R. Brown and S. G. Colt respectively.

Engines *G*, *H* & *K* are at the new plant of the Cambridge Electric Light Company, and were tested by the writer.

Description of the Engines.

3. The engines tested vary from 750 to 2,500 rated horse power, and were all built by McIntosh, Seymour & Co., of Auburn, N. Y. They are all vertical, high-speed, two-cylinder, cross-compound, direct-connected units with overhanging cranks, Each cylinder is supported on a heavy, hollow, cast-iron frame at the back and on two inclined steel standards in front. Each H. P. cylinders is jacketed on the barrel, and both heads and the jackets are piped in series; the steam enters the jacket on the top head, passes into the barrel jacket, goes to the jacket on the lower head and then to the reheater coils. In this way a very active circulation in the jackets is ensured. As there is no separate steam supply to the reheater coils, nor any separate drain from the H. P. jackets, it is not possible to use either jackets or reheater alone. The receiver is a large cylindrical drum at the back of the engine and close to the cylinders. The reheater consists of one or more coils of pipe in the receiver. The L. P. cylinder is unjacketed.

4. The valves are of the flat, gridiron type, unbalanced and of short stroke. The steam valves on both H. P. and L. P. cylinders consist of a main valve cutting off at about .8 stroke, and a Rider cut-off valve, the movement of which can be varied so as to give any desired cut-off. The main steam valves and the exhaust valves on each cylinder are driven from an eccentric on the main shaft through a system of links and levers. The cut-off valves are driven by auxiliary eccentrics which are controlled by a fly-wheel governor. The action of the valves is rapid; the openings for admission and exhaust of steam are large.

The fly-wheel governors are designed to control the speed within two per cent. variation between zero load and full load. At the Cambridge electric light station the position of the governor weights can be regulated when the engine is running by

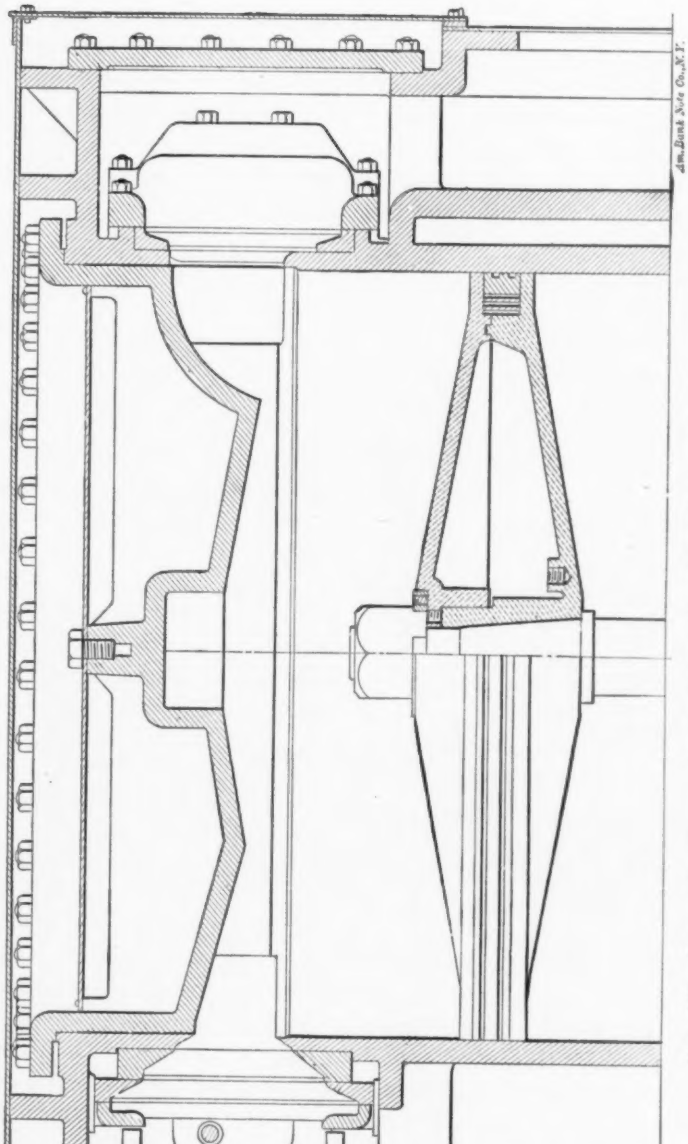


FIG. 117.—SECTIONAL ELEVATION OF UPPER HALF OF 60″ × 56″ L. P. CYLINDER.

means of a small electric motor fastened to the fly wheel and controlled from the switch-board. This device is valuable for synchronising in parallel running.

The Arrangements for Testing.

5. All the engines, with one exception to be noted later, are fitted with jet condensers so that their steam consumptions had

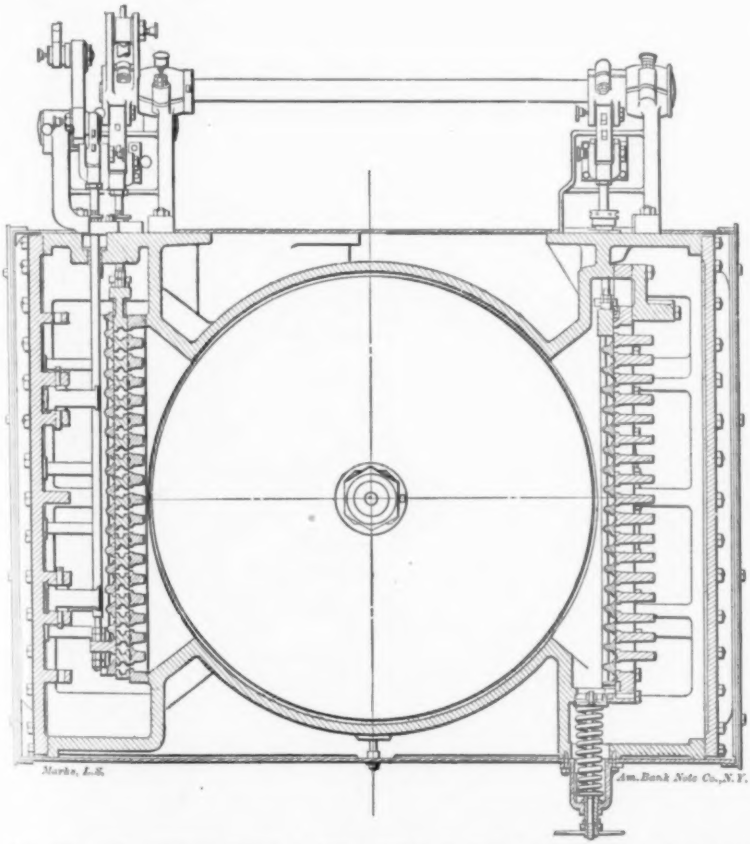


FIG. 117A.—SECTIONAL PLAN OF 60″ × 56″ L. P. CYLINDER SHOWING ADMISSION AND EXHAUST VALVES.

to be determined by measuring the boiler feed. This necessitated the adoption of adequate precautions to prevent leakage from the feed mains, from the boilers and from the steam pipes.

The steam was obtained in every case from Babcock & Wilcox boilers, fed from a pump which handled only the water going to the test boilers. These boilers supplied steam only to the engine under test; they were examined for general tightness;

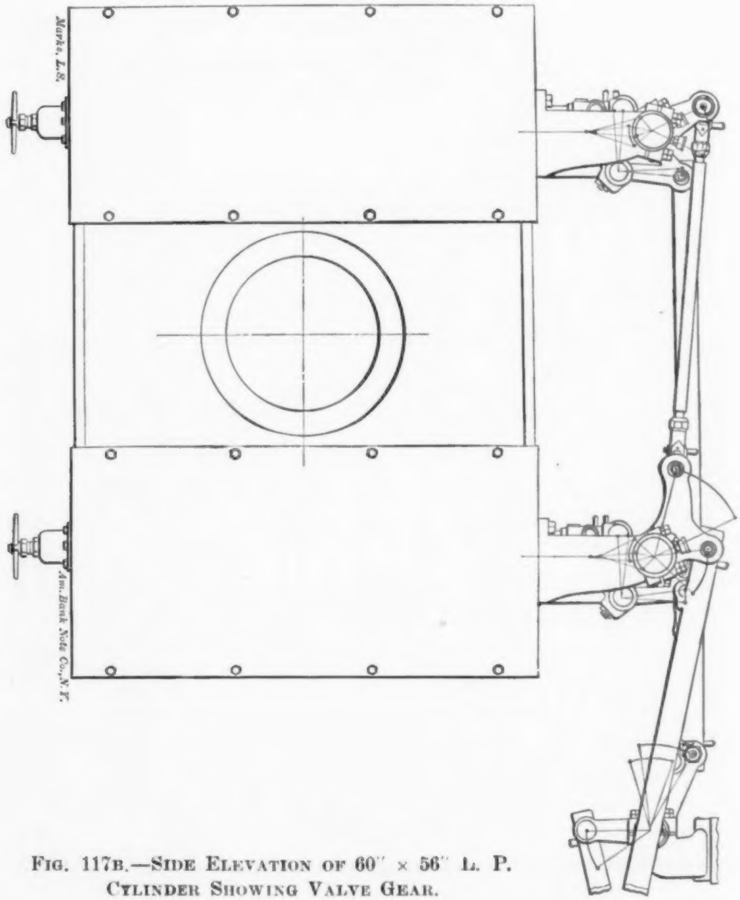


FIG. 117B.—SIDE ELEVATION OF 60″ × 56″ L. P. CYLINDER SHOWING VALVE GEAR.

were shut off from the Holly return systems and had their blow-offs blanked. The steam on its way to the engine under test went first to a section of the main steam header, which was isolated by gate valves from the rest of the header. These gate valves were tested for tightness with full steam pressure on one side and atmospheric pressure on the other, and were

found practically tight in all cases. In order, however, to prevent even slight leakage the steam pressures were kept the same on both sides of the valves during the tests. In most of the tests the steam supply to the engines was superheated so that the steam pipe drips could be closed; in the cases where wet steam was supplied the condensation in the pipes was returned by gravity to the test boilers.

6. The total leakage from the boilers and steam pipes was determined by leakage tests after nearly every run. To make the leakage test, the fires were allowed to burn down at the end of the run and were kept in such condition as just to be able to maintain the steam at the test pressure when the supply to the engine was shut off completely. The observation of the rate of lowering of the water level in the boilers under these conditions gave the necessary information for determining the rate of leakage loss from the boilers and steam pipe.

7. The feed heaters were all tested and found to be tight. The reheater coils were also found to be tight except in one engine and in that case the only run made was without the reheater in use.

8. The weight of steam condensed in the jackets and reheater, when these were in use, was determined by collecting the condensed steam in a vessel of known capacity provided with a gauge glass. The drainage from the receiver was determined in a similar way. The arrangement of jackets and reheater prevented the separate determination of the amounts of condensation occurring in each.

9. The diameters of all the cylinders were gauged when hot. The clearances were not measured directly; the values used for the calculations were given by the engine builders and were determined by them from the working drawings.

10. The weighing scales were examined by the local sealer of weights, and their accuracy in the writer's tests was ensured by further proving with special test weights. All gauges, thermometers and indicators were calibrated. The indicators used in the tests of engine *A* were calibrated under steam pressure by comparison with a mercury column at the Crosby Steam Gage Company's works, and later were tested at the Engineering Laboratory of Harvard University under steam pressure by comparison with known rotating weights. The results of these two calibrations are given below.

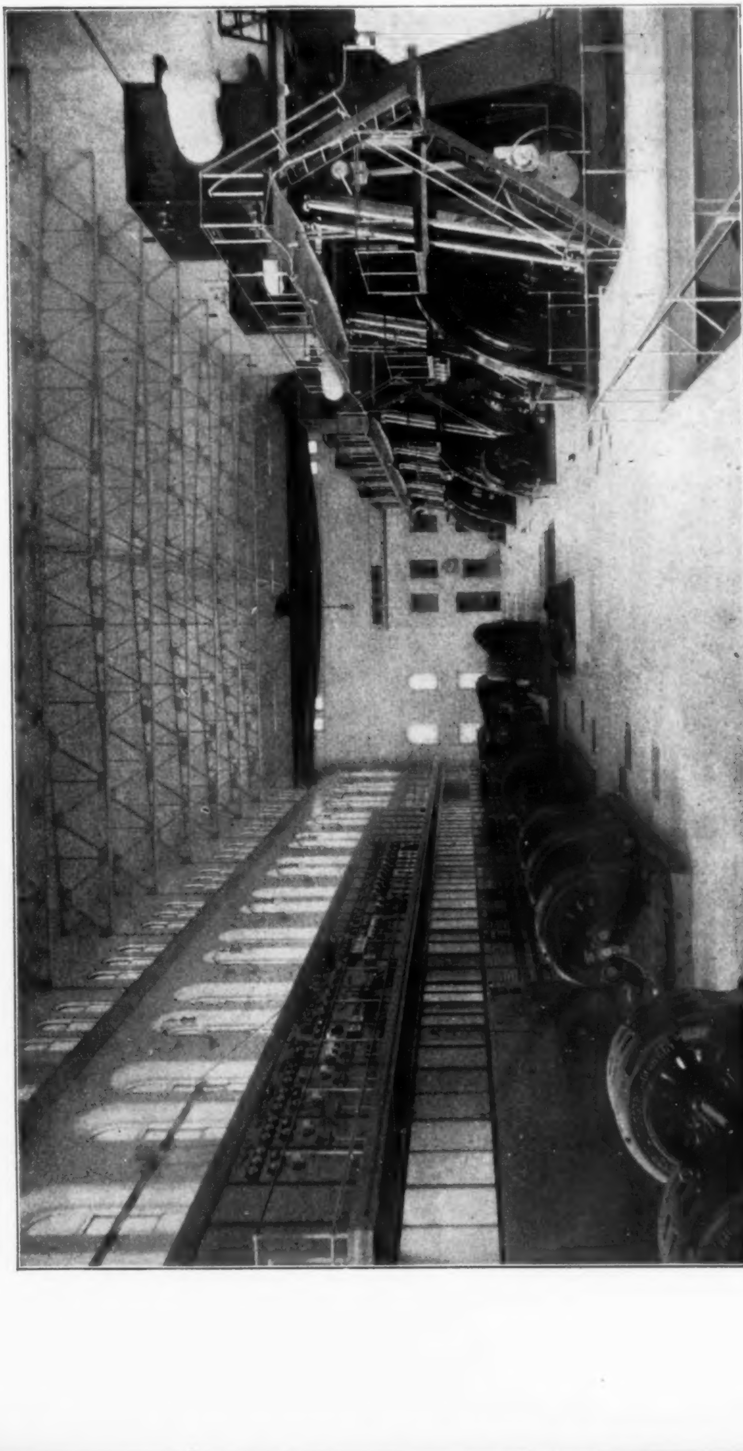


FIG. 118.—ENGINE ROOM OF THE L. ST. STATION OF THE BOSTON ELECTRIC LIGHT CO. SHOWING ENGINE A.

Maker's Name.	Number of Instrument.	Nominal Scale of Spring.	Actual Scale of Spring.	
			By Mercury Column.	By Rotating Weights.
Tabor	783	100	98.9	98.3
Tabor	1926	100	99.7	99.4
Crosby	4908	20	19.48	19.4
Crosby	4909	20	19.5	19.57

11. The load on the engine was entirely electrical, and consisted of part of the station load supplemented when necessary by an adjustable water rheostat load. There was no difficulty in any of the tests in keeping the total load constant.

Tests of Engine A.

12. In June, 1899, the writer, assisted by students in engineering of Harvard University, made the acceptance test of the engines then installed at the L Street station of the Boston Electric Light Company. A description of the plant may be found in the "Electrical World and Engineer," for May 27, 1899. Of the three similar units then installed, engine *A* (number 1 of the plant) was selected for test because it had been running longer than the others.

13. This engine is rated to develop 2,400 indicated horse-power at .23 cut-off, 4,128 indicated horse-power at .6 cut-off and has a maximum cut-off at .8 stroke. Its piston speed is 960 feet per minute; its cylinder dimensions 28 inches and 58 inches by 48 inches. It is direct connected to a 1,500 kilowatt, three-phase General Electric generator delivering current at 2,250 volts. The fly wheel is 16 feet diameter and weighs 100,000 pounds. The main shaft is 26 inches diameter; the bearings 21 inches by 42 inches. The steam pipe is 10 inches, the exhaust 24 inches diameter. The ratio of low pressure to high pressure piston displacements is 4.3 to 1. The reheater coil has 777 square feet of heating surface, but the steam supply to it was too small.

14. The engine was apparently in first class condition; the valves and pistons were tight when at rest. The steam was obtained from two 550 H. P. boilers, which supplied saturated steam to the engine. The amount of feed water was determined by direct weighing.

15. Two full load tests were made: one with the jackets and

reheater in use, the other with both out of use. The principal results of the test are given in Table I. The combined indicator cards for the two tests are given in Figs. 119 and 120. A comparison of the two tests shows the conditions to have been very similar during the two runs with some advantage in the matter of vacuum for the first test. The reheater was not very effective owing to the deficient steam supply—it superheated the receiver steam 35 degrees Fahr.

TABLE I.

GENERAL RESULTS OF TESTS ON ENGINE A.

1	Number of test	1	2
2	Nominal load	Full	Full
3	With or without jackets and reheaters	With	Without
4	Duration of test, hours	10	5
5	H. P. cylinder diameter, inches	28.14	28.14
6	L. P. " " "	58.12	58.12
7	Piston rod diameter, inches	5.5	5.5
8	Clearance H. P. cylinder, per cent	3.6	3.6
9	" L. P. " "	6.35	6.35
10	Ratio L. P. to H. P. displacement	4.3	4.3
11	Revolutions per minute	118.46	118.46
12	Gauge pressure at throttle	159	161
13	Steam qaulity at throttle, per cent	99.46	99.5
14	Initial steam pressure in H. P. cylinder, by cards	156	155
15	Steam pressure in receiver, gauge	21	20
16	Superheat of steam entering L. P. cylinder, Fahr	35.5	0
17	Effective vacuum in L. P. cylinder, by cards, inches	24.05	23.05
18	Vacuum in exhaust pipe, inches	25.0	24.5
19	Temperature of exhaust steam, Fahr	127	132
20	Commercial cut-off H. P. cylinder, head end	.29	.295
21	" " " " crank end	.30	.305
22	" " L. P. " head end	.29	.32
23	" " " " crank end	.31	.3[illegible]5
24	I. H. P., H. P. cylinder	859	961.1
25	" L. P. "	1,135	1,022.6
26	Total I. H. P.	1,994	1,983.7
27	Percentage power developed by H. P. cylinder	43	48.5
28	Total dry steam per I. H. P. per hour, pounds	13.59	13.68
29	Wet steam in jackets and reheater per I. H. P. per hour	.92	0
30	Percentage of total steam used in jackets and reheater	6.8	0
31	Percentage of cylinder steam drained from receiver	1.6	6.0
32	Quality at cut-off, H. P. cylinder	82	76
33	Quality at release, H. P. cylinder	86	80
34	Quality at cut-off, L. P. cylinder	92.5	87
35	Quality at release, L. P. cylinder	95.5	93
36	B. T. U. per I.H.P. per minute, measured above ideal feed temperature	245	248
37	Thermodynamic efficiency, per cent	17.3	17.1
38	B. T. U. per I. H. P. per minute, for Rankine ideal cycle	160	161
39	Thermodynamic efficiency compared with ideal cycle, per cent	65.4	65
40	Percentage saving by jacketing and reheating	1.3	

16. The economic results as indicated by the steam consumptions in line 28 show practically no saving at all by the use of the high pressure jackets and the reheater. The steam consumption, however, is not a fair quantity by which to judge the performance of the engine, since in Test 1 about 7 per cent. of the total steam is rejected as condensation at about the boiler temperature, whereas in Test 2 all is rejected at the condenser

temperature. Since, in these tests, engine performances alone are being studied, and the efficiencies of the feed heaters are not considered at all, the performance of the engine is best measured by the amount of heat that must be given to the feed water per I. H. P. per minute, the feed being supplied at what may be known as the ideal feed temperature. The ideal feed temper-

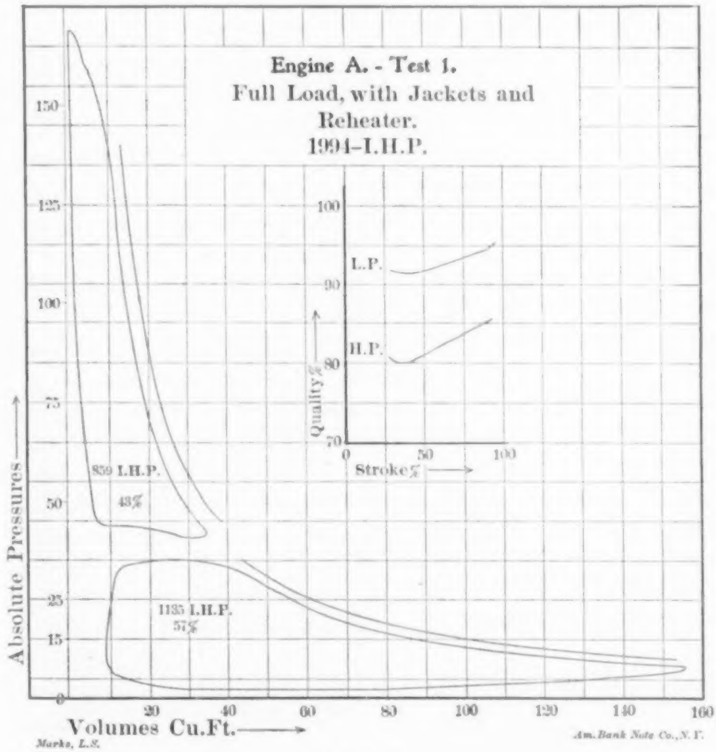


FIG. 119.

ature is obtained by mixing the condensations in the reheater and receiver, each at its own proper temperature, with the condensation water in the condenser supposed to be at the exhaust pipe temperature. The heat units per I. H. P., measured above the ideal feed temperature, necessary to form steam such as exists at the throttle, is an accurate measure of the performance of the engine, and it is this quantity which is given in line 36, and from which the calculation in line 40 of the saving by jacket-

ing and reheating has been made. An examination of the combined cards in Figs. 119 and 120, or lines 32 and 33 of Table 1, show that the H. P. jackets have been moderately effective, raising the quality of the steam in the H. P. cylinder 6 per cent. throughout the expansion. The small total gain by both jackets and reheater indicate then that the reheater is of small value,

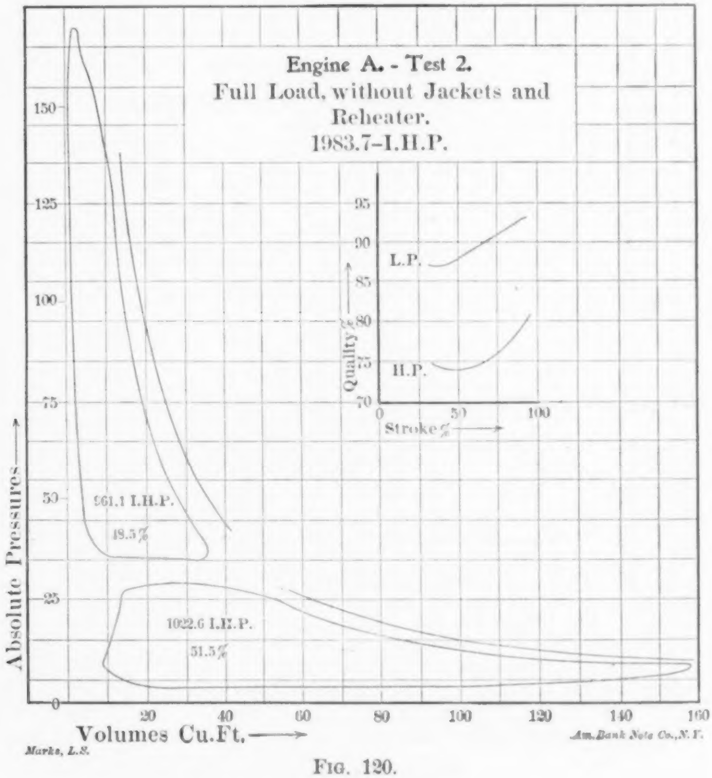


FIG. 120.

or may even be a source of loss when it superheats only 35 degrees.

Tests of Engine B.

17. The engines *B*, *C*, *D*, *E* and *F* are all part of the Atlantic Avenue plant of the Edison Electric Illuminating Company, the station numbers of these engines being 7, 8, 9, 10 and 11 respectively. This plant has been described in the "Electrical World

and Engineer," of May 18, 1901, and also in the paper by Messrs. Moultrop & Curtis, published in Volume XXIII. of the *Transactions* of this Society. The engines *C*, *D*, *E* and *F* are all of the same size and are rated at about 2,400 indicated horse power. Engine *B* develops only about one-half the power of the other engines; it is also the only one of these engines fitted with a surface condenser.

18. Owing to the comparatively short duration of test necessary when the steam consumption of an engine is measured by weighing the discharge from a surface condenser, it was practicable to carry out a much more complete series of tests with this engine than with any of the others. The engine *B* is much more distant from the boilers than the other engines tested, and in consequence the superheat at the throttle is very low, and drop of pressure in the steam pipe is considerable.

19. The engine is intended to run at 100 revolutions per minute and to develop 1,200 indicated horse-power at .22 cut-off with 160 pounds steam pressure and 26 inches vacuum. At .6 cut-off it develops 2,200 indicated horse-power. The cylinder dimensions are 23 inches and 48 inches by 48 inches, and the engine drives an 800 kilowatt General Electric direct-current generator. The reheater consists of coils of brass pipe aggregating 440 square feet total heating surface. The high pressure clearance is 2 per cent.; the low pressure is given as 3 per cent. The diameter of the fly wheel is 15 feet and its weight 65,000 pounds. The main bearings are 17 inches diameter, 35 inches long; the diameter of the shaft between the bearings 20 inches. The diameters of the steam and exhaust pipes are 9 inches and 20 inches respectively.

20. Ten separate tests were made on this engine in order to determine its economy at different loads, both with and without the jackets and reheater in use. Of these tests, five (viz.: tests 3, 4, 6, 8 and 10) were with the jackets in use, and were with loads increasing from one-quarter load to an overload of one-quarter. Four tests (viz.: tests 5, 7, 9 and 11) were without the use of jackets and reheater and were with loads varying from one-half to an overload of one-quarter. Another test (number 12) was made at full load with jackets and reheater in use and with a greater boiler pressure than in the previous runs so as to test the engine more nearly under its rated conditions.

21. The principal results of these ten tests are given in Table

TABLE II.

General Results of Tests on Engine B.

1	Number of test	3	4	5	6	7	8	9	10	11	12
2	Nominal load	¼	½	½	¾	¾	1.0	1.0	1¼	1¼	1.0
3	With or without jackets and reheater	With	With	Without	With	Without	With	Without	With	Without	With
4	Duration of test, hours	2	2	2	2	2	5	2	1	1	8
5	Revolutions per minute	103.6	103.4	103.1	102.5	102.5	101.93	101.9	101.2	101.2	101.97
6	Gauge pressure at throttle	152.6	151.5	155.3	150	155.1	149.5	158.9	152.6	155.2	161.8
7	Superheat of steam at throttle, Fahr.	.8	4.8	21.8	10.7	17.5	9.2	14.6	8.0	13.8	21.6
8	Initial steam pressure in H. P. cylinder, from cards	97.8	130.05	133.8	131.6	133.7	130.0	134.8	131.2	135.9	140.18
9	Steam pressure in receiver, gauge	1.33	8.8	7.0	13.8	11.0	16.5	13.8	....	19.7	20.0
10	Superheat of steam to L. P. cylinder, degrees Fahr.	99.8	75.4	Wet	60.5	Wet	46.6	Wet	52.9	Wet	49.4
11	Effective vacuum, L. P. cylinder, from cards	26.7	26.0	25.8	26.5	25.5	25.85	25.93	24.6	23.56	25.66
12	Vacuum in exhaust pipe	28.22	28.0	27.25	28.3	27.08	27.71	28.0	25.2	26.82	27.0
13	I. H. P., H. P. cylinder	121.0	203.0	247.0	316.0	388.0	413.0	507.0	608.0	702.0	415.0
14	I. H. P., L. P. cylinder	188.0	349.0	301.0	529.0	443.0	634.0	598.0	890.0	814.0	689.0
15	Total I. H. P.	309.0	552.0	548.0	845.0	831.0	1,047.0	1,105.0	1,498.0	1,516.0	1,104.0
16	Percentage power developed in H. P. cylinder	39.4	36.8	45.1	37.4	46.7	39.5	45.9	40.6	46.3	37.6
17	Electrical output, K. W.	154.9	328.8	330.0	537.4	535.2	674.0	731.5	981.8	1,004.6	707.6
18	Efficiency of unit $= \frac{\text{E. H. P.}}{\text{I. H. P.}}$	67.3	80.0	83.2	85.2	85.8	86.3	88.8	88.0	88.8	82.0
19	Total steam used per I. H. P. per hour, pounds	14.04	13.71	15.11	13.63	14.72	13.59	14.41	14.57	15.16	13.58
20	Steam used in jackets and reheater per I. H. P. per hour, pounds	1.432	1.225	....	1.08	....	0.947	....	0.908	....	0.940
21	Percentage steam used in jackets and reheater	10.2	8.95	....	7.91	....	7.49	....	8.9	....	6.90
22	Percentage cylinder steam drained from receiver	....	....	3.7	....	6.70	....	5.97	....	4.97	....
23	Quality at cut-off, H. P. cylinder	61.0	58.0	60.0	68.0	72.0	68.0	60.0	79.0	74.0	65.0
24	Quality at release, H. P. cylinder	85.0	85.0	88.0	80.0	74.0	78.0	75.0	83.0	78.0	81.0
25	Quality at cut-off, L. P. cylinder	95.0	90.0	65.0	88.0	78.0	90.0	76.0	94.0	88.0	90.0
26	Quality at release, L. P. cylinder	95.0	95.0	91.0	88.0	90.0	91.0	83.0	98.0	89.0	93.0
27	B. T. U. per I. H. P. per minute, measured above exhaust temperature	264	257	279	255	272	252	269	261	267	246
28	B. T. U. per I. H. P. per minute, measured above ideal feed temperature	253	252	277	251	270	248	267	255	265	241
29	B. T. U. per I. H. P. per minute, measured above ideal feed temperature, with effective vacuum = 26 inches	258	252	274	251	266	247	267	250	257	240
30	Thermodynamic efficiency	16.5	16.9	15.4	17.0	15.8	17.2	15.90	16.7	16.1	17.6
31	B. T. U. per I. H. P. per minute, Rankine cycle	150	150	160	152	160	155	154	168	163	155
32	Thermodynamic efficiency compared to ideal engine	58	60	58	60.5	59	62.5	58	66.1	61.5	64.5
33	Percentage saved by jacketing and reheating	....	9.3	....	7.5	....	5.6	....	4.0	....	....

II. The combined indicator cards are shown in Figs. 6 to 15. On examining the conditions under which these tests were made it will be seen that the superheat was slight in all cases, but was

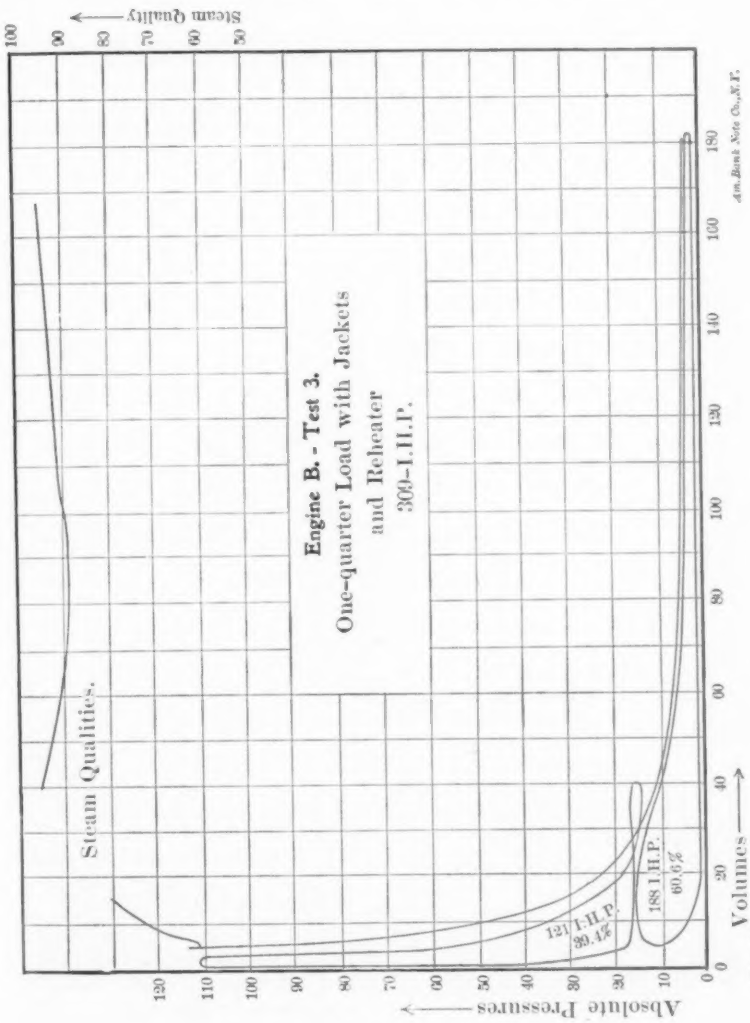


Fig. 121.

greater during the series of tests without jackets and reheater. The vacuum was not constant, but fell at the higher loads; it averaged higher than in the tests of any of the other engines. In order to eliminate the effect of variation in the vacuum on

the economy of the engine a correction has been applied reducing all the results to an effective vacuum of 26 inches. This correction is obtained by adding to the mean effective pressure

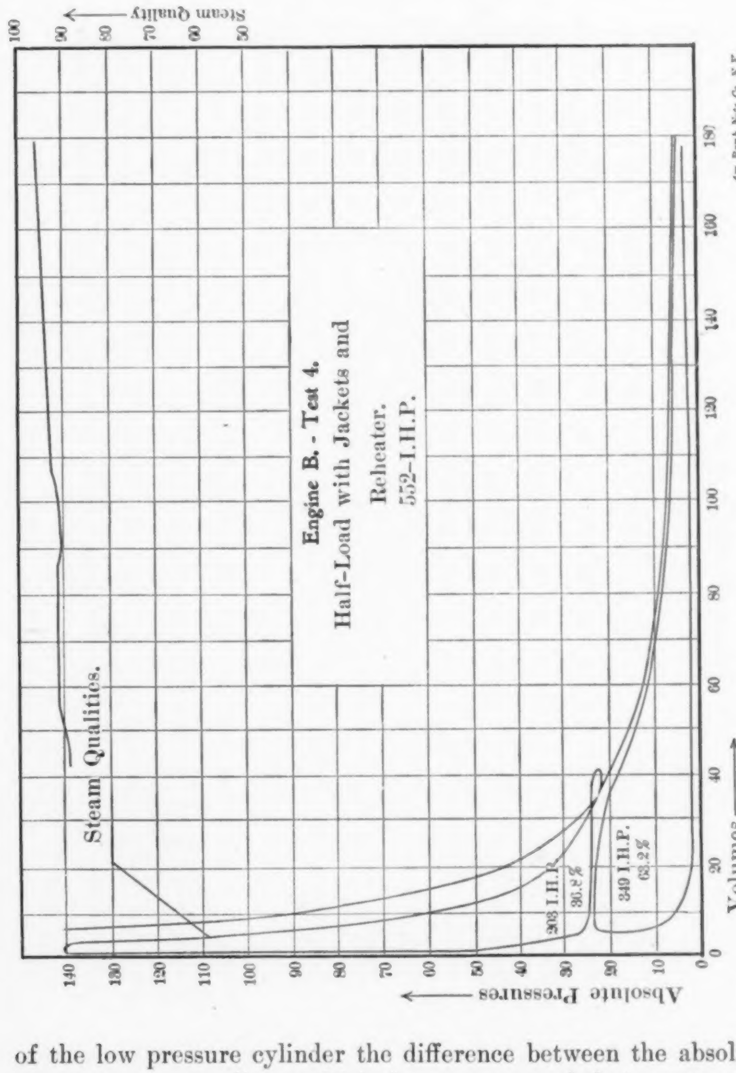


Fig. 122.

of the low pressure cylinder the difference between the absolute pressure corresponding to 26 inches vacuum and that corresponding to the actual effective vacuum shown by the cards and recorded in line 11. The corrected results are given in line 29,

and for the purpose of comparison of results the figures in this line are the most satisfactory to use, since they not only eliminate the efficiency of the feed water heaters but also reduce all results to a common vacuum.

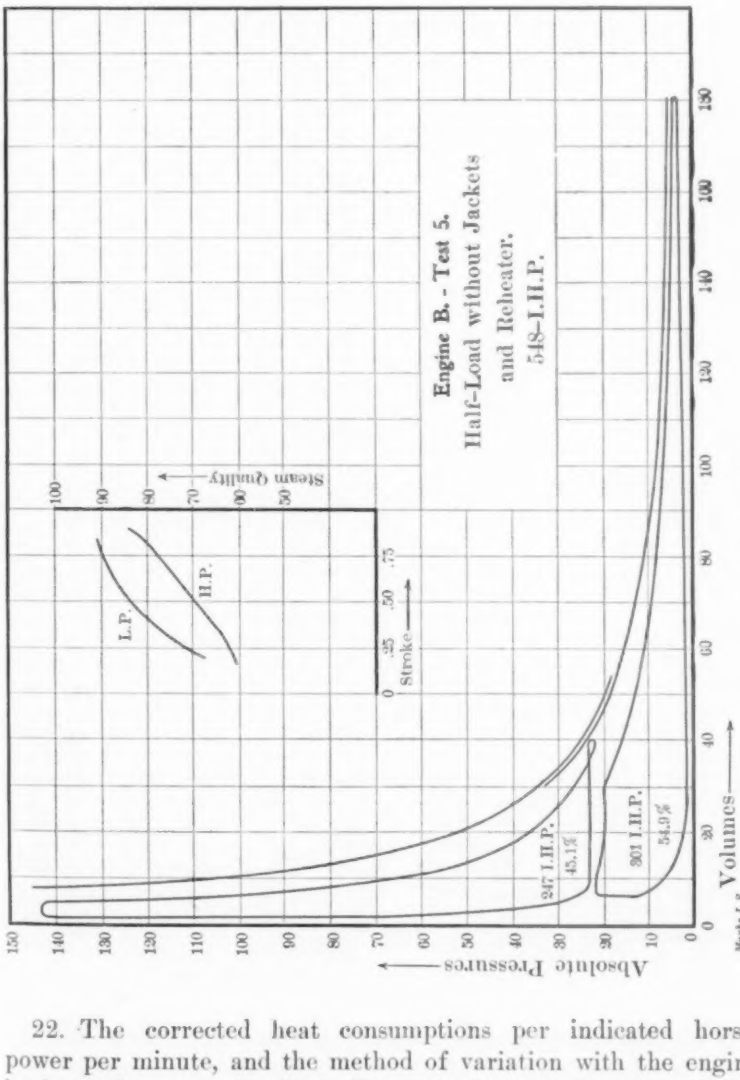


Fig. 123.

22. The corrected heat consumptions per indicated horsepower per minute, and the method of variation with the engine load, are shown graphically in Fig. 131; the same figure shows also the percentage of the total steam used in the jackets and re-

and for the purpose of comparison of results the figures in this line are the most satisfactory to use, since they not only eliminate the efficiency of the feed water heaters but also reduce all results to a common vacuum.

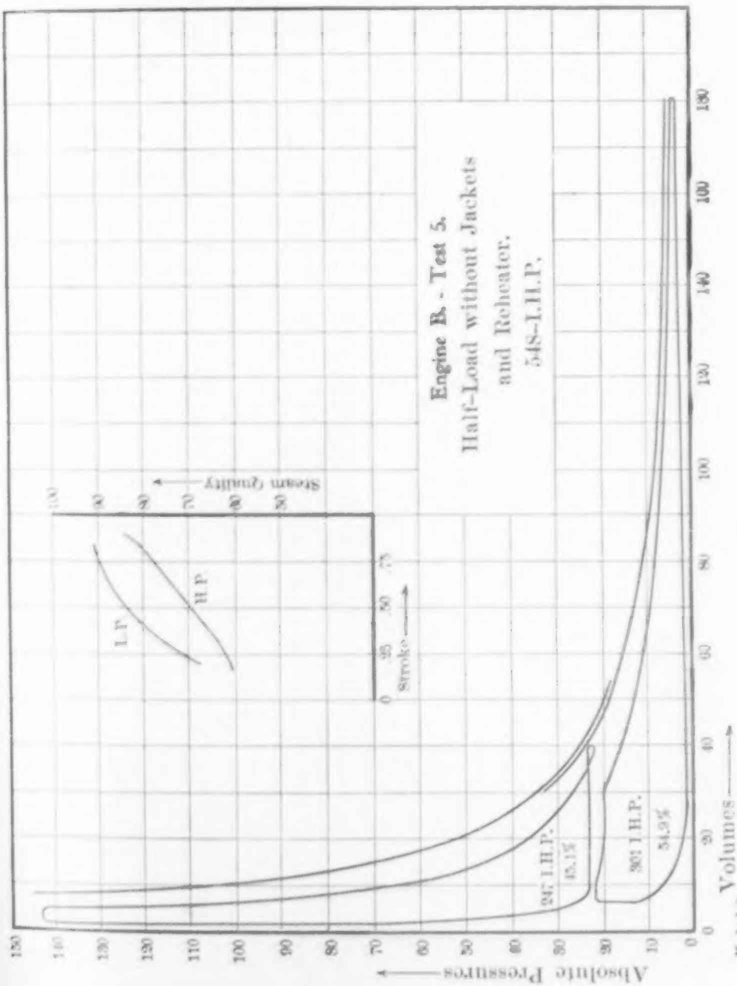


Fig. 123.

22. The corrected heat consumptions per indicated horsepower per minute, and the method of variation with the engine load, are shown graphically in Fig. 131; the same figure shows also the percentage of the total steam used in the jackets and re-

heater. The most striking feature of the curves is the small variation of the efficiency of the engine when jackets and reheater are used throughout the range of load from one-half load to one-quarter overload. Within this range the total variation of econ-

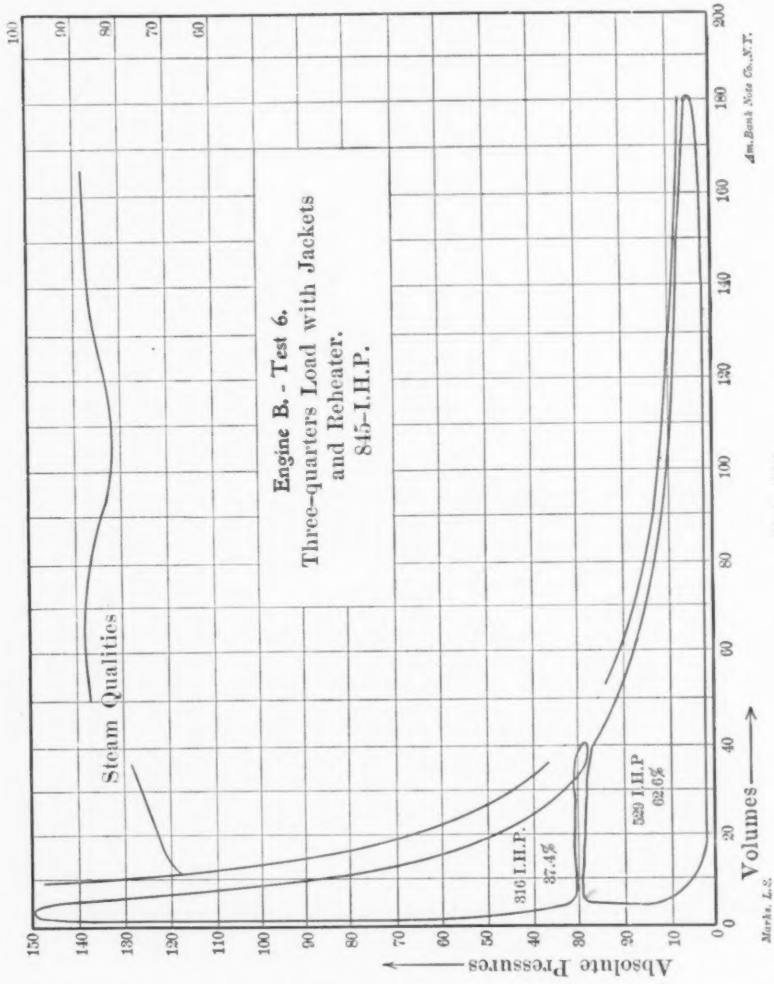


FIG. 124.

omy did not exceed 2 per cent., and the economy was greatest at about the rated load. It should also be noted that the percentage of the total steam used in the jacket and reheaters is lowest when the efficiency is greatest.

23. Without the jackets and reheater in use the economy increases with increase of load, and in such way as to indicate that there would be but little advantage in the use of jackets and reheater when the engine is much overloaded. With light loads

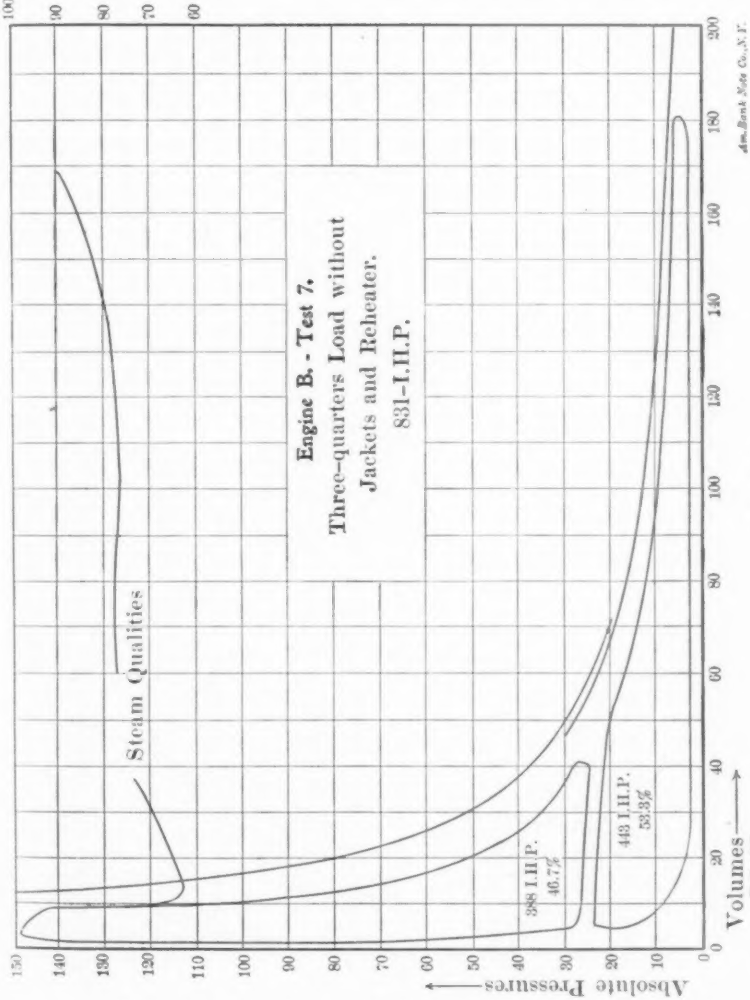


FIG. 125.

up to full load the saving by jackets and reheater ranges from 9 to 4 per cent. and is obtained by using from 10 to 7 per cent. of the total steam in the jackets and reheater. The increase in economy with the load and the reheater in use would have been

greater had the reheater been of greater capacity. As is seen in line 10, the superheat of the steam entering the low pressure cylinder decreased from 100 degrees to 46 degrees as the amount of

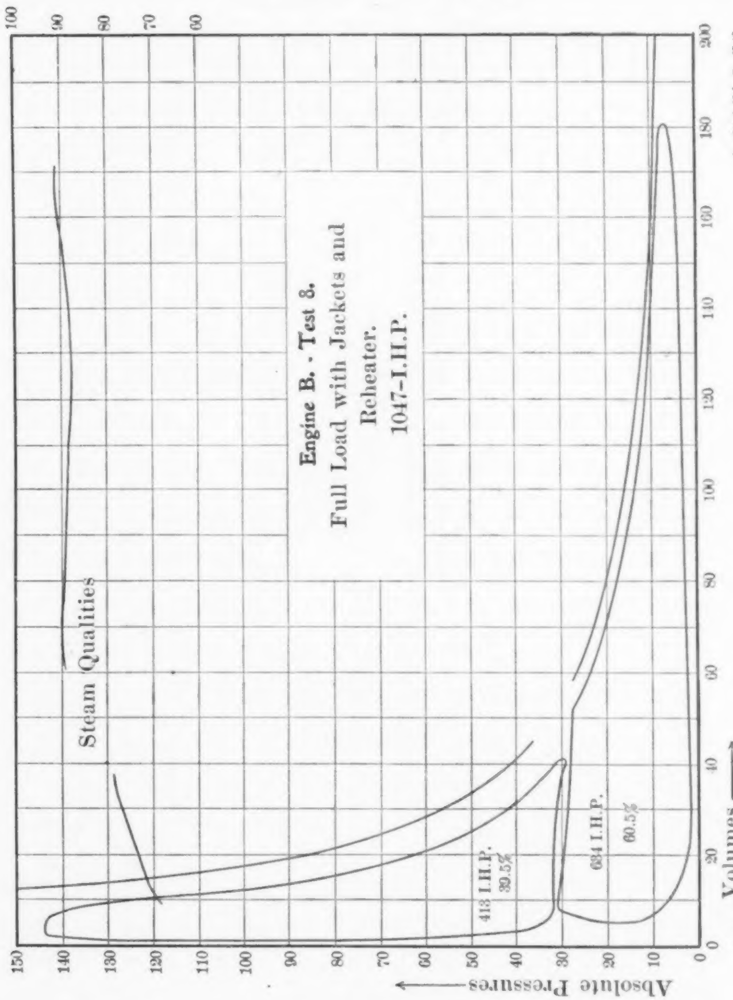


FIG. 126.

steam used increased. This is probably the reason for the close approach of the two heat consumption curves in Fig. 130. An examination of the indicator cards, Figs. 120 to 129, and of the steam qualities tabulated in lines 23 to 26 of Table II., shows that the qualities during expansion in the high pressure cylinder

are but slightly affected by the steam jackets. On the other hand, the qualities during expansion in the low pressure cylinder show very clearly the beneficial results of reheating, especially

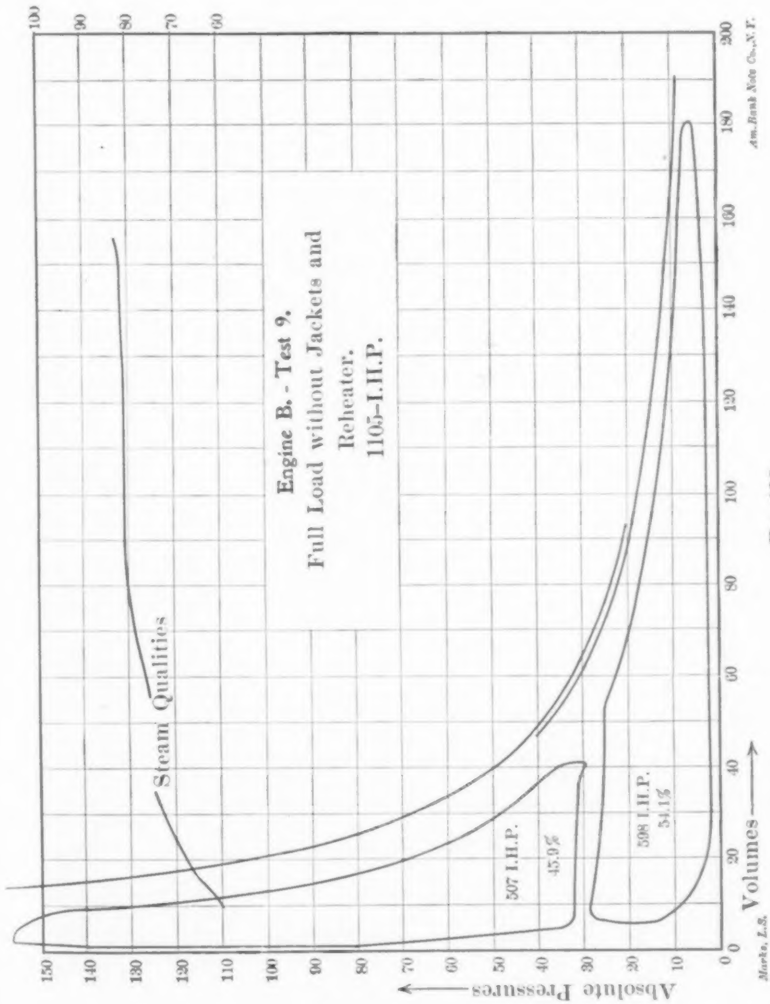


FIG 127.

in the low load tests where the superheat in the receiver was greatest. The rise of the end of the expansion line of the high pressure cards at light loads, Figs. 121, 122 and 123 results from the return of steam from the receiver to the high pressure cylinder

due to the lifting of the exhaust valve from its seat when the pressure in the cylinder becomes less than that in the receiver.

Tests of Engines C, D, E and F.

24. The engines *C, D, E* and *F* are all located in the new engine room of the Atlantic Avenue station of the Edison

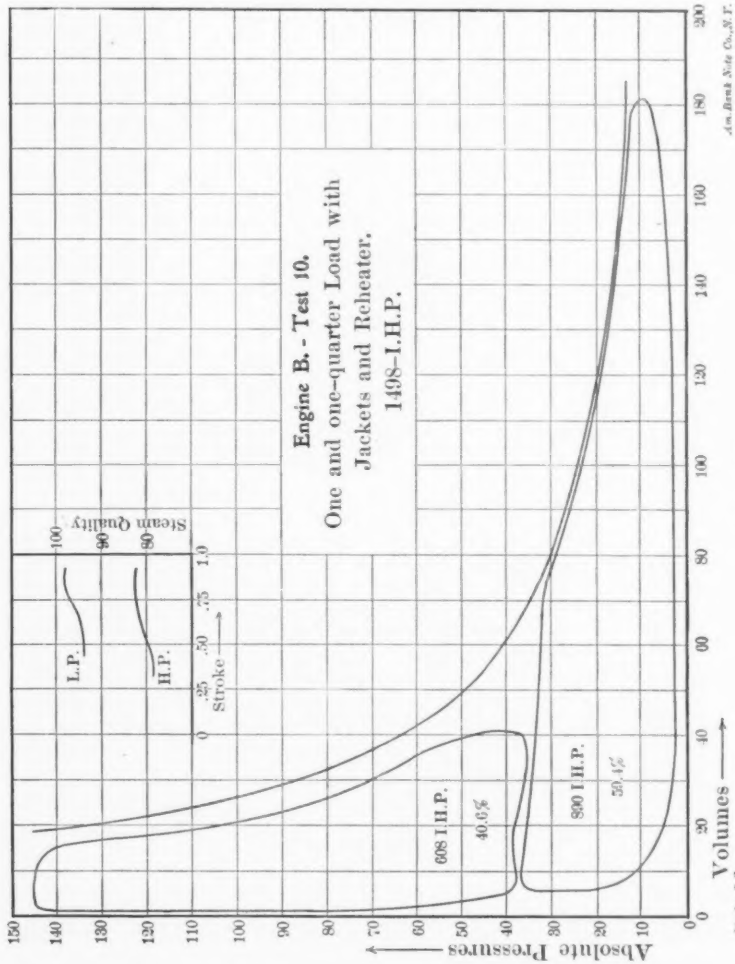


Fig. 128.

Electric Illuminating Company, and are almost identically similar units, each rated at about 2,400 indicated horse-power and

direct connected to a 1,600 kilowatt direct current General Electric generator. The cylinders are all 29 inches and 60 inches by 56 inches; the fly wheels 18 feet diameter and 130,000 pounds

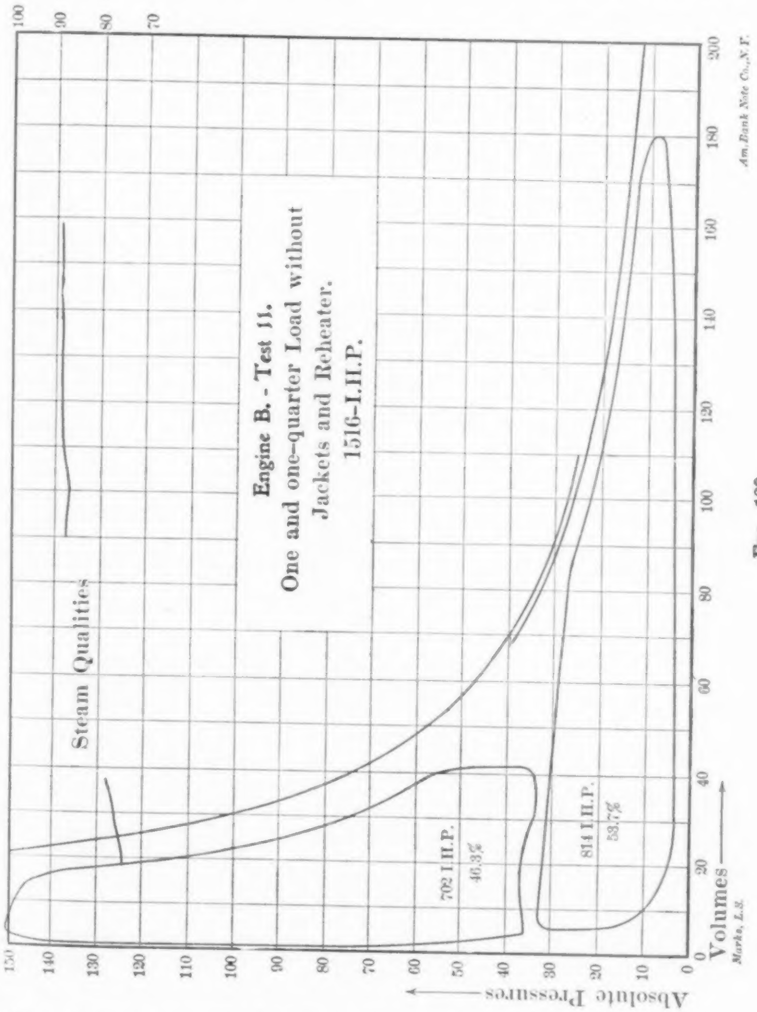


FIG. 129

in weight; the main bearings 24 inches diameter and 48 inches long; the shafts between the bearings 27 inches diameter; the steam and exhaust pipes 10 inches and 24 inches diameter respectively. The high pressure clearances are 2.75 per cent.; the low pressure clearances 4 per cent.

25. Engines *C* and *D* have each 740 square feet of reheating surface in the receiver; the other two engines have each 800 square feet. The only other differences between these four

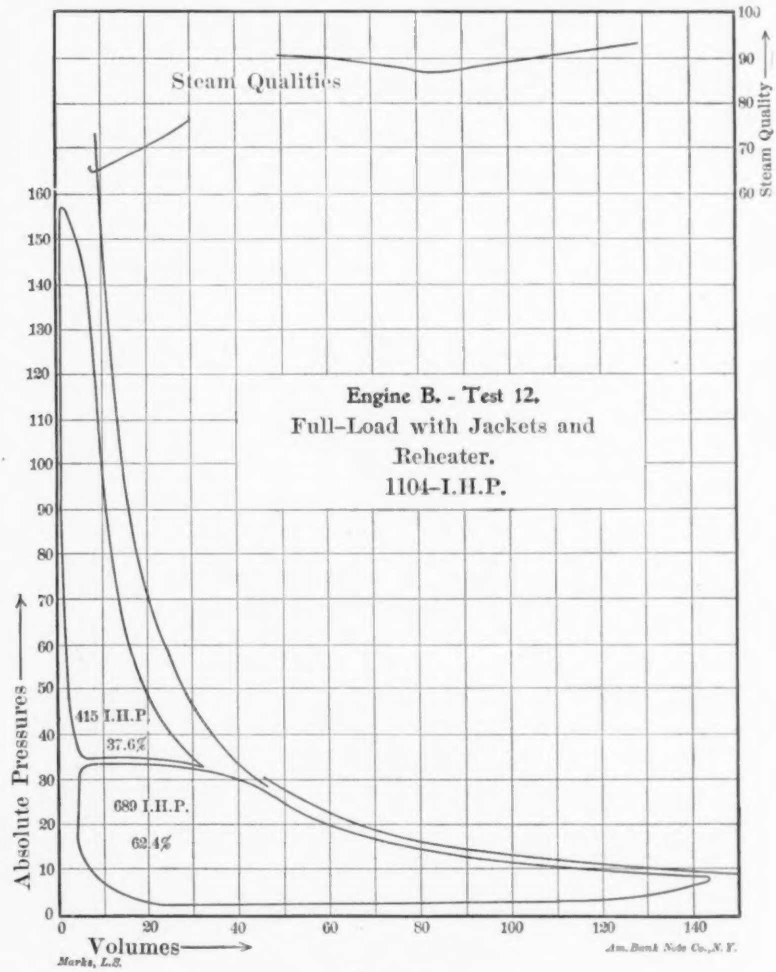


FIG. 130.

engines are minor differences in the valve gearing. Engines *C* and *D* were tested in 1901; engines *E* and *F* in the following year. All these four engines exhaust into a central condensing plant consisting of jet condensers and Blake twin vertical air pumps.

The measurement of the feed-water was accomplished by the use of 2-inch Worthington hot water meters, one at each boiler employed during the list. These meters were calibrated by sending known weights of water through them at each of

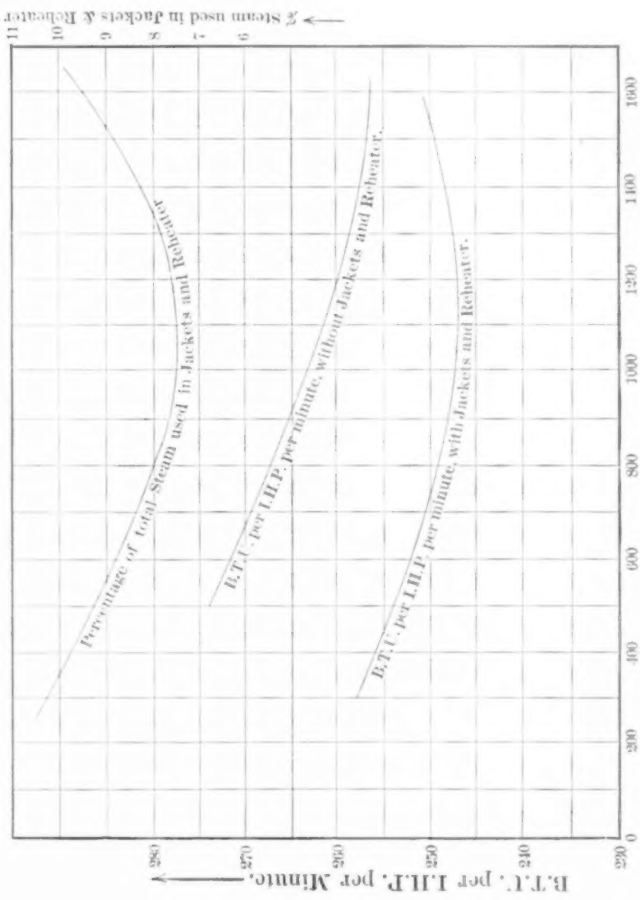


FIG. 131.—RESULTS OF TESTS OF ENGINE B.

three different speeds. The middle speed in each case was approximately the average speed of working of the meters during the test. The meters showed errors which were practically constant throughout the range of speeds employed, so that their readings could be used with complete confidence. Boiler leakage tests were made as usual after each run.

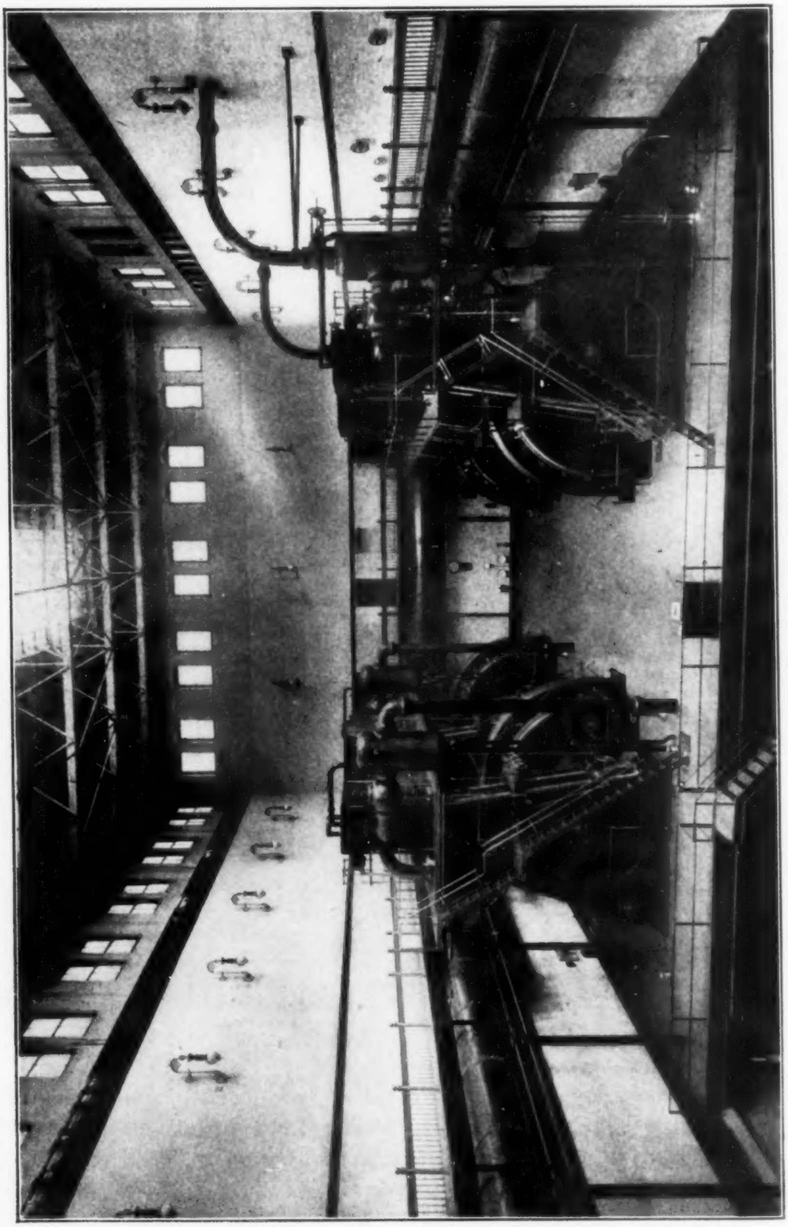


FIG. 132.—ENGINE ROOM AT THE ATLANTIC AV. STATION OF THE EDISON ELECTRIC ILLUMINATING CO., SHOWING ENGINES C, D, E, AND F.

26. The following tests were made on these engines:

Engine *C* was tested at full, three-quarters and half loads with jackets and reheater in use, and at full load without steam in the jackets and reheater.

Engine *D* was tested at half load and at full load, both with and without steam in the jacket and receiver, but as a drip

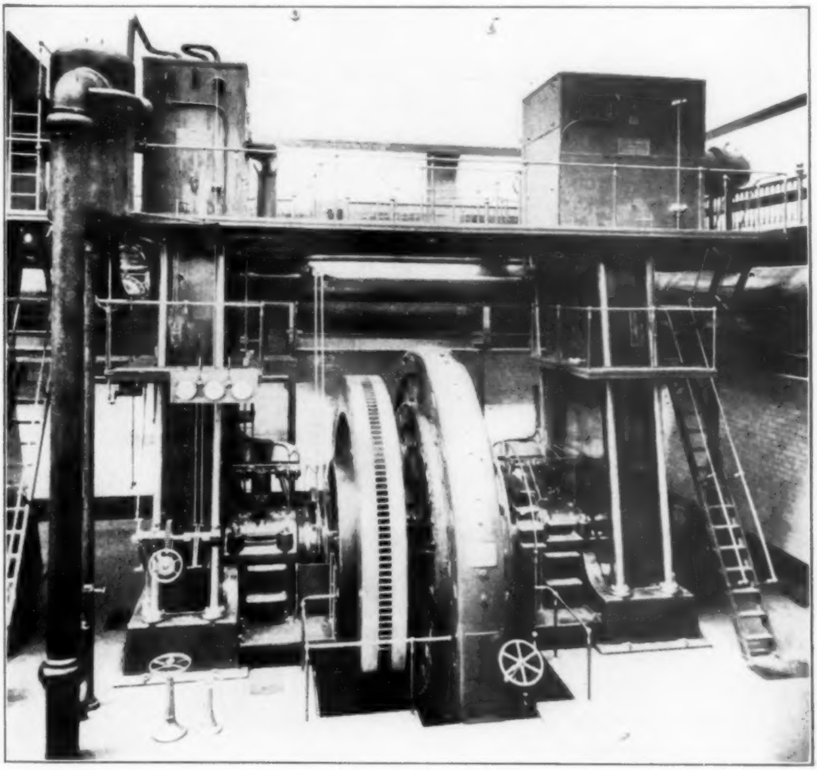


FIG. 133.—ENGINE C.

connection, which should have been closed, was found to be partly opened after the two half-load tests, the results of these tests are not given here, and the table of results contains only those obtained at full load.

Engine *E* was tested at full and at half load with jackets and reheater in use.

Engine *F* was tested only at full load, with steam in the jackets and reheater.

27. The principal r sults of these nine tests are collected in Table III. The combined indicator cards for the same tests are shown in Figs. 134 to 142.

28. The tests on engine C show the efficiency of the engine to change very slightly with change of load,—the better economy at half load being principally due to the better vacuum, though this is offset in part by the lower superheat of the admission steam.

The saving by jacketing and reheating at full load is only 3 per cent. The jacketing is shown by the steam qualities in the high pressure cylinder to be of no value with 80 degrees admission superheat, and the reheater is of but small value when it superheats only 60 degrees.

29. The economic results of engine D are better than on engine C as a result of higher superheat of the admission steam and a better vacuum. The effect of the higher superheat is most plainly shown by the very high steam qualities during expansion in the high pressure cylinder. The saving in this engine by the use of jackets and reheater is 4.5 per cent.—a little greater than in engine C. The full load tests on engines E and F gives results agreeing very closely with those obtained on engine D—the running conditions were very similar in all three cases. The half load test on engine E gives the same economic result as the full load test on the same engine.

Tests of Engines G, H and K.

30. These engines are part of the new plant of the Cambridge Electric Light Company, and correspond to station numbers 1, 2 and 3 respectively. A description of the plant is to be found in the "Engineering Record" for November 1, 1902.

31. Engines G and H are exactly similar units, 18 inches and 38 inches by 42 inches, developing 760 indicated horse-power at .24 cut-off with 135 pounds initial steam pressure and 26 inches effective vacuum, and direct connected to 600 kilowatt, 60-cycle alternating generators built by the General Electric Company. Engine K has cylinder dimensions 31 inches and 64 inches by 48 inches, develops 2,320 indicated horse-power at .24 cut-off with 135 pounds initial steam pressure and 26 inches effective vacuum and is direct connected to a 1,500 kilowatt General Electric generator.

TABLE III.

General Results of Tests on Engines C, D, E, and F.

1	Engine	C	C	C	C	D	D	E	E	F
2	Number of test	**13**	**14**	**15**	**16**	**17**	**18**	**19**	**20**	**21**
3	Nominal load	1	1	3/4	1/2	1	1	1	1/2	1
4	With or without jackets and reheater	With	Without	With	With	With	Without	With	With	With
5	Duration of test, hours	10	5	4	4	8	8	8	6	8
6	Revolutions per minute	97.48	97.64	98.77	99.6	98.18	98.61	99.72	100.69	98.53
7	Gauge pressure at throttle	168.6	166.7	167	169.4	162.8	163.1	159.2	160.4	158.7
8	Superheat at throttle, degrees Fahr.	80.2	78.5	64.5	48.5	98.4	98.4	72.4	70.4	92.7
9	Initial steam pressure in H. P. cylinder, from cards	158.1	155.5	155.4	150.5	157	157.3	156.0	153.3	150.3
10	Steam pressure in receiver, gauge	13.75	11.1	10.1	5.5	13.5	12.3	14.0	7.5	14.5
11	Superheat of steam entering, L. P. cylinder, Fahr.	60.3	....	67.0	84.8	60.5	....	60.9	86.0	61.7
12	Effective vacuum in L. P. cylinder from cards, inches	23.32	23.00	23.95	24.33	24.3	24.38	24.75	24.4	24.75
13	Vacuum in exhaust pipe, inches	24.95	24.71	25.59	26.14	25.55	25.66	25.34	25.45	26.08
14	I. H. P., H. P. cylinder	1,098.4	1,217.8	874	572.7	1,082.5	1,189	1,119.1	632.9	1,008.4
15	I. H. P., L. P. cylinder	1,168.6	1,022	957	666.2	1,119.1	1,026.1	1,111.0	626.0	1,162.7
16	Total I. H. P.	2,267.	2,239.8	1,831	1,238.9	2,201.6	2,215.1	2,230.1	1,258.9	2,171.1
17	Percentage power developed by H. P. cylinder	48.5	54.3	47.8	46.2	49.1	53.8	50.2	50.3	46.5
18	**Efficiency of unit** $\frac{E.\ H.\ P.}{I.\ H.\ P.}$	91.	90.5	90.8	87.8	91.0	......	90.5	86.1	91.0
19	Total steam per I. H. P. per hour, pounds	12.72	12.98	12.71	12.38	11.57	11.88	11.67	11.78	11.67
20	Steam in jackets and reheater per I. H. P. per hour, pounds	.825	....	.915	1.15	.85	....	.830	1.09	.655
21	Percentage steam used in jackets and reheater	6.5	....	7.2	9.3	7.4	....	7.1	9.3	5.6
22	Percentage of cylinder steam drained from receiver	....	2.3	....	....	....	....	....	....	....
23	Quality at cut-off, H. P. cylinder	86	86	77	65	99	93	93	82	89
24	Quality at release, H. P. cylinder	83	88	82	80	94	87	94	92	84
25	Quality at cut-off, L. P. cylinder	99	84	93	99.8	100	86	93	85	100
26	Quality at release, L. P. cylinder	96	91	98.5	99	96	84	93	92	100
27	B. T. U. per I. H. P. per minute measured above ideal feed temperature	236	244	235	227	217	226.5	217	217	221
28	Thermodynamic efficiency	18.0	17.4	18.1	18.7	19.6	18.7	19.6	19.6	19.2
29	B. T. U. per I. H. P. per minute, Rankine cycle	161	160	158	154	164	164	158	158	157
30	Thermodynamic efficiency compared to ideal engine	68.3	65.6	67.3	67.7	75.4	72.3	72.6	73.	70.7
31	Percentage saved by jacketing and reheating	3.0	....	....	....	4.5	....	....	....	....

32. A slight leak was found in the reheater coils of engine H, so that this engine was run only without steam in the reheater. Engine K had been used but very little, and had not worn down

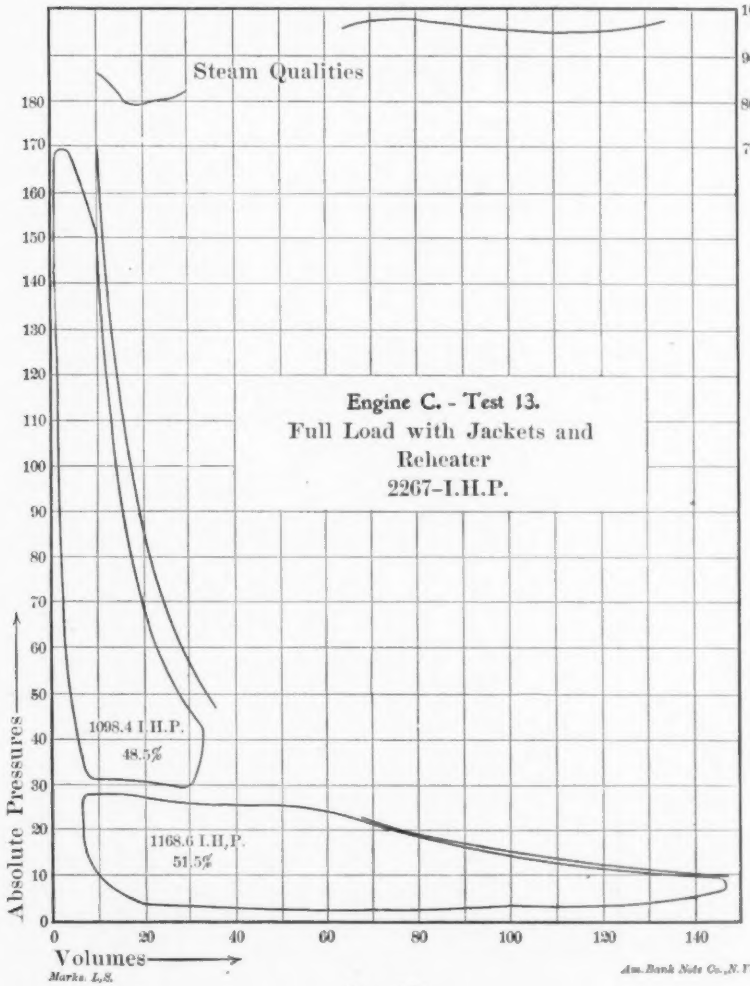


FIG. 134.

to such good condition as the other two engines. Its governor was found to stick at very light loads, and the consequent hunting made it impracticable to obtain satisfactory friction cards from the engine when running without load. In all other respects

the conditions were favorable for satisfactory tests of all three engines.

33. During the tests the whole of the steam generated in the

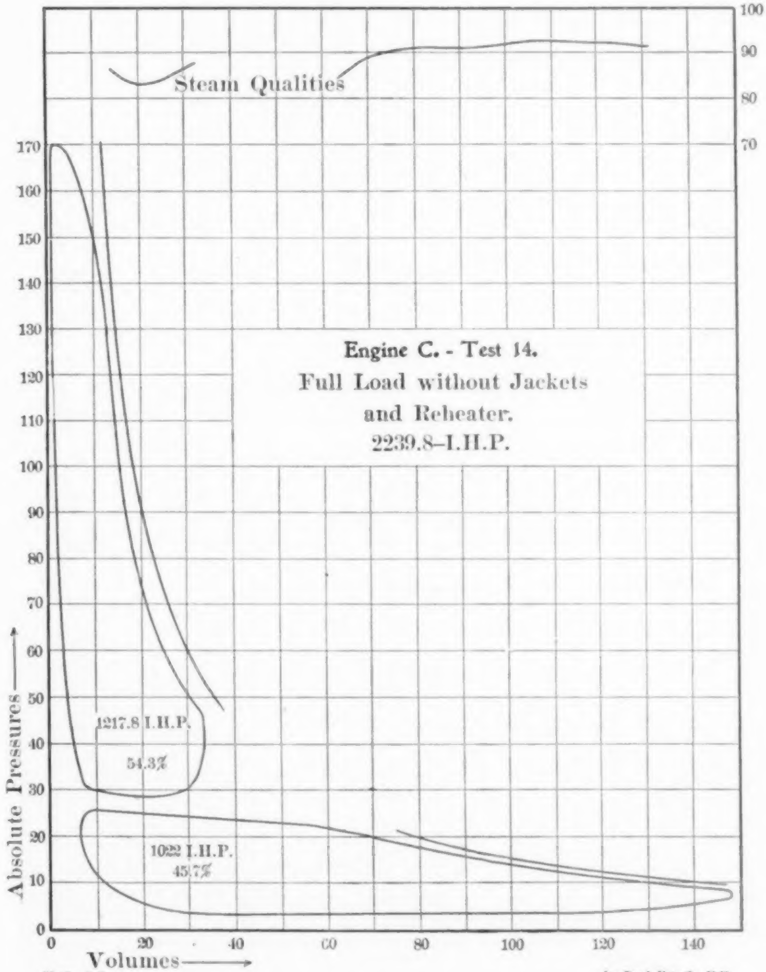


FIG. 135.

plant was used for the engine under test and its auxiliaries. The main engine was supplied from one or more main boilers; the auxiliaries from an auxiliary boiler. The feed to the main boilers was determined by direct weighing. The engine under

test carried the total station load, and to this was added the power absorbed by a water rheostat, continuously adjusted so as to keep the sum of the loads constant. The tests were made in

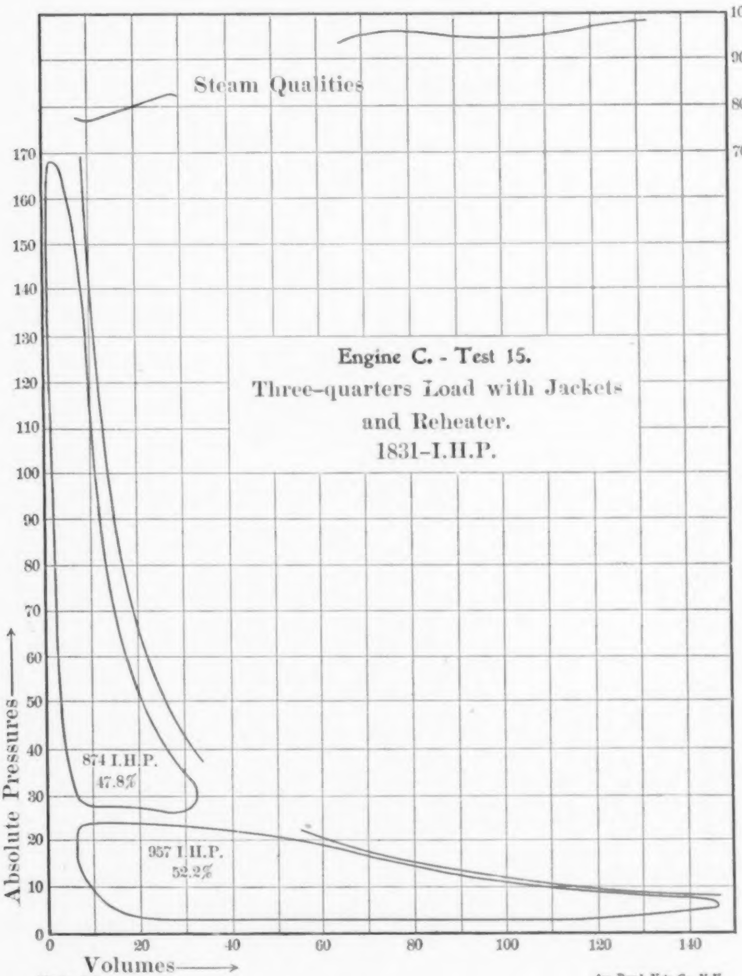


FIG. 136.

April and May, 1903, and were conducted by the writer, assisted by students in engineering of Harvard University. On engine *G* two full load tests were made, one with, the other without steam admission to the jackets and reheater. Only one test

was made on engine *H;* a test with about 20 per cent. overload and without steam in the jackets and reheater. Engine *K* was tested at full, half and quarter loads with the jackets and reheater in use, and at half load without them.

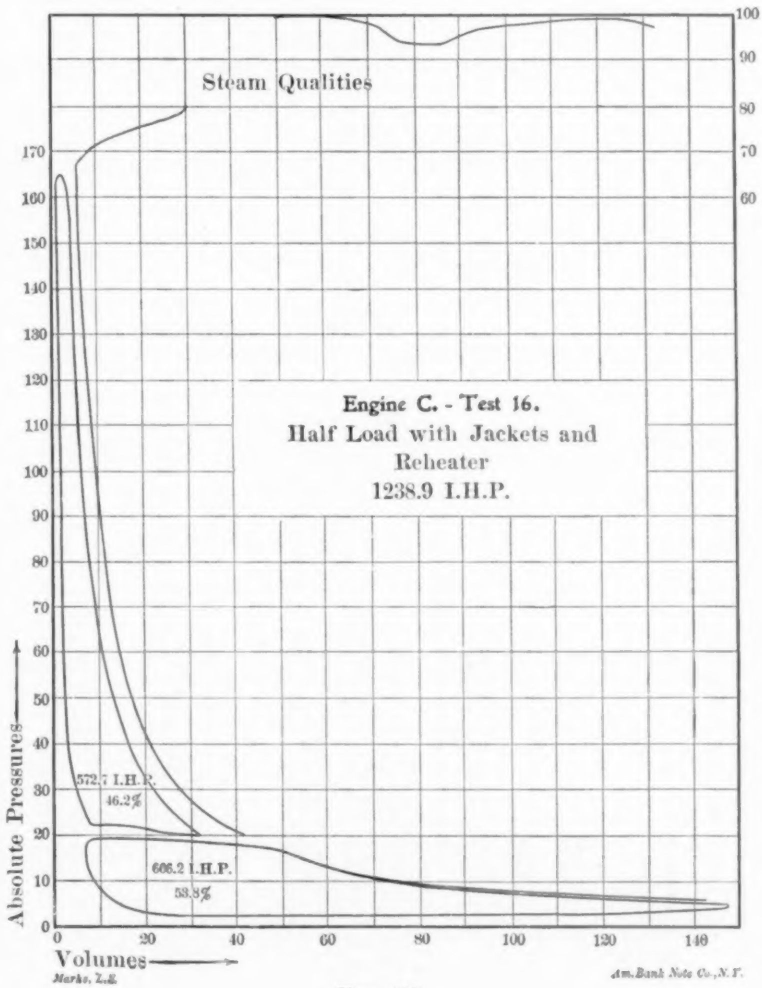


FIG. 137.

34. The principal results of the tests on engines *G*, *H* and *K* are collected in Table IV. The combined indicator cards for the seven tests are given in Figs. 143 to 149.

TABLE IV.

GENERAL RESULTS OF TESTS ON ENGINES G, H, AND K.

1	Number of test	22	23	24	25	26	27	28
2	Engine	G	G	H	K	K	K	K
3	Nominal load	1	1	1¼	1	½	¼	½
4	With or without jackets and reheater	Without	With	Without	With	With	With	Without
5	Duration of test, hours	10	10	9	10	8	8	8
6	H. P. cylinder, diameter, inches	18.00	18.00	18.024	31.037	31.037	31.037	31.037
7	L. P. cylinder diameter, inches	38.03	38.03	38.05	64.035	64.035	64.035	64.035
8	Piston rod diameters, inches	5	5	5	5.5	5.5	5.5	5.5
9	Clearance H. P. cylinder, per cent	4.9	4.9	4.9	6.1	6.1	6.1	6.1
10	Clearance L. P. cylinder, per cent	3.95	3.95	3.95	8.1	8.1	8.1	8.1
11	Ratio L. P. to H. P. piston displacements	4.59	4.59	4.59	4.32	4.32	4.32	4.32
12	Revolutions per minute	118.4	119.1	119.15	116.9	119.11	119.34	118.37
13	Gauge pressure at throttle, pounds per square inch	137.8	139.5	133	138.5	134	137.8	136.5
14	Superheat at boiler, degrees Fahr	88.2	91	74.3	82.6	89.5	81.0	92.3
15	Superheat at throttle, degrees Fahr	72.2	75	55.5	64.2	70.2	55.3	73.5
16	Initial steam pressure in H. P. cylinder, by cards, pounds per square inch	130.9	136.9	130	132.9	122.4	95	126.1
17	Absolute steam pressure in receiver, pounds per square inch	23.6	25.8	27.8	25.3	19.8	16.2	18.3
18	Superheat of steam entering L. P. cylinders, degrees Fahr	....	49.1	....	40.3	59	79.1	....
19	Effective vacuum in L. P. cylinder, by cards, pounds per square inch	11.82	12.17	11.8	11.96	12.43	12.54	12.22
20	Vacuum in exhaust pipes, inches	26.15	26.27	26	25.4	26.4	26.4	25.8
21	Temperature of exhaust steam, degrees Fahr	125.7	125.2	133.9	133.7	125.4	121.5	129
22	Commercial cut-off H. P. cylinder, head end	.285	.360	.363	.217	.063	.069	.074
23	Commercial cut-off H. P. cylinder, crank end	.266	.225	.331	.187	.071	.050	.095
24	Commercial cut-off L. P. cylinder, head end	.338	.288	.331	.269	.145	.10	.118
25	Commercial cut-off L. P. cylinder, crank end	.354	.303	.329	.248	.162	.112	.157
26	Commercial ratio of expansion	16.81	19.28	13.64	21.4	54.3	68.9	49.9
27	M. E. P., H. P. cylinder, pounds per square inch	62.74	56.19	74.60	54.95	29.19	15.93	29.3
28	M. E. P., L. P. cylinder, pounds per square inch	12.26	13.55	14.52	12.30	6.38	4.15	5.98
29	Equivalent M. E. P. referred to L. P. cylinder, pounds per square inch	25.29	25.16	30.11	24.03	12.92	7.79	13.03
30	I. H. P., H. P. cylinder	383.9	347.1	463.3	1,063.8	604.7	330.8	646.8
31	I. H. P., L. P. cylinder	330.8	368.4	394.3	1,120.2	591.2	385.2	551.3
32	Total I. H. P.	714.7	715.5	857.6	2,184	1,195.9	716	1,198.1
33	Per cent. total power developed in H. P. cylinder	53.7	48.6	54.0	48.8	51.0	46.2	54.5
34	Diagram factor	.74	.85	.77	.87	.90	.83	.82
35	Total steam per I. H. P. per hour, pounds	14.13	13.21	13.13	12.50	12.217	12.95	12.90
36	Steam in jackets and reheater per I. H. P. per hour, pounds	....	.85	....	.61	.89	1.17	....
37	Per cent. of total steam used in jackets and reheater	....	6.45	....	4.89	7.28	9.05	....
38	Per cent. of cylinder steam drained from receiver	3.35	.69	6.12	.93	.67	....	2.75
39	Quality of steam at cut-off, H. P. cylinder, per cent	76	85	86	92	83	81	80
40	Quality of steam at release, H. P. cylinder, per cent	75	88.5	83	93	82.5	88	92
41	Quality of steam at cut-off, L. P. cylinder, per cent	84	89	84	93	92	97.4	80.5
42	Quality of steam at release, L. P. cylinder, per cent	87	93.2	82	97	96	100	94
43	Calculated ideal feed water temperature, degrees Fahr	133	144	137	162.5	143.3	143	131.6
44	Heat per pound of steam above ideal feed water temperature, B. T. U	1,124.8	1,115.7	1,111.3	1,091.8	1,112.8	1,106.6	1,126.5
45	B. T. U. per I. H. P. per minute above ideal feed water temperature	264.9	245.5	243.3	227.3	226.6	241.5	244.0
46	Thermodynamic efficiency	16.01	17.34	17.43	18.7	18.72	17.6	17.4
47	B. T. U. per I. H. P. per minute for Rankine's ideal cycle	166.3	165	167.5	168.6	166	161.3	164.7
48	Thermodynamic efficiency compared with ideal cycle, per cent	62.8	67.4	68.8	74.2	73.3	66.8	67.5
49	Per cent. saving by jacketing and reheating	....	7.5	....	....	7.2	....	....

35. The engines G and H are the smallest of the whole series tested. The results of the two full load tests of G agree very closely with the results of the tests 9 and 8 of engine B under

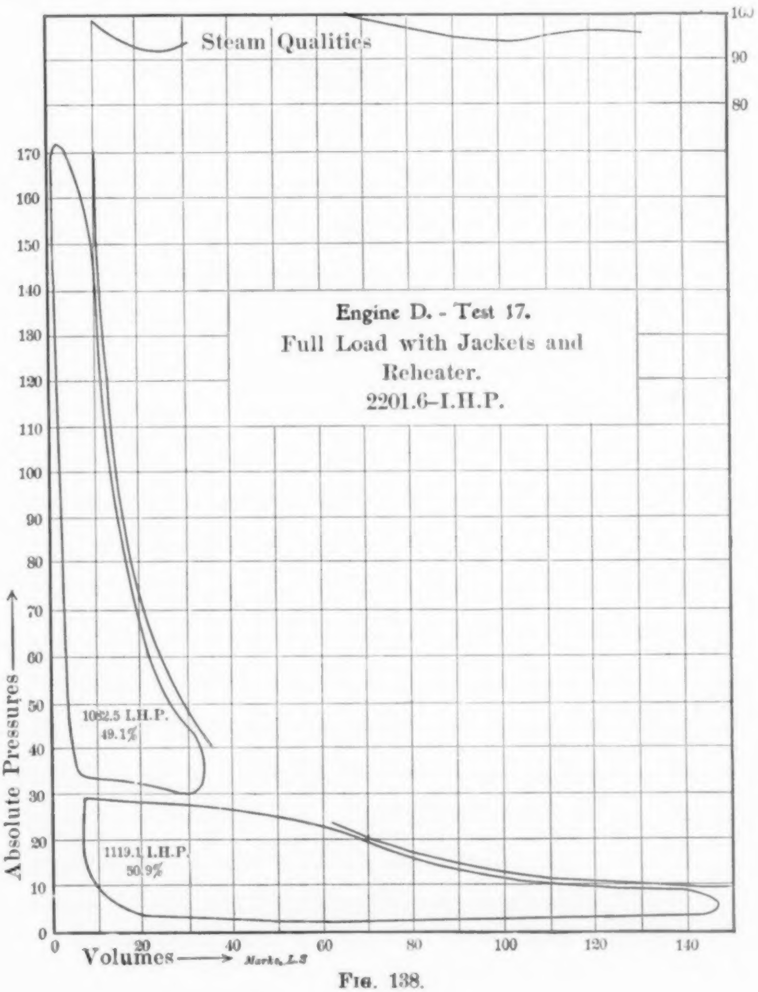


Fig. 138.

full load, although this latter engine is 50 per cent. larger than G. The reheaters gave practically the same superheat in both cases; the steam pressures in the high pressure cylinders were practically the same, so that the disadvantage of smaller size and poorer vacuum in engine G is just about offset by the greater

superheat of its admission steam. The gain by the use of jackets and reheater was about 7 per cent. in both cases. It will be seen by an examination of the steam qualities in the high press-

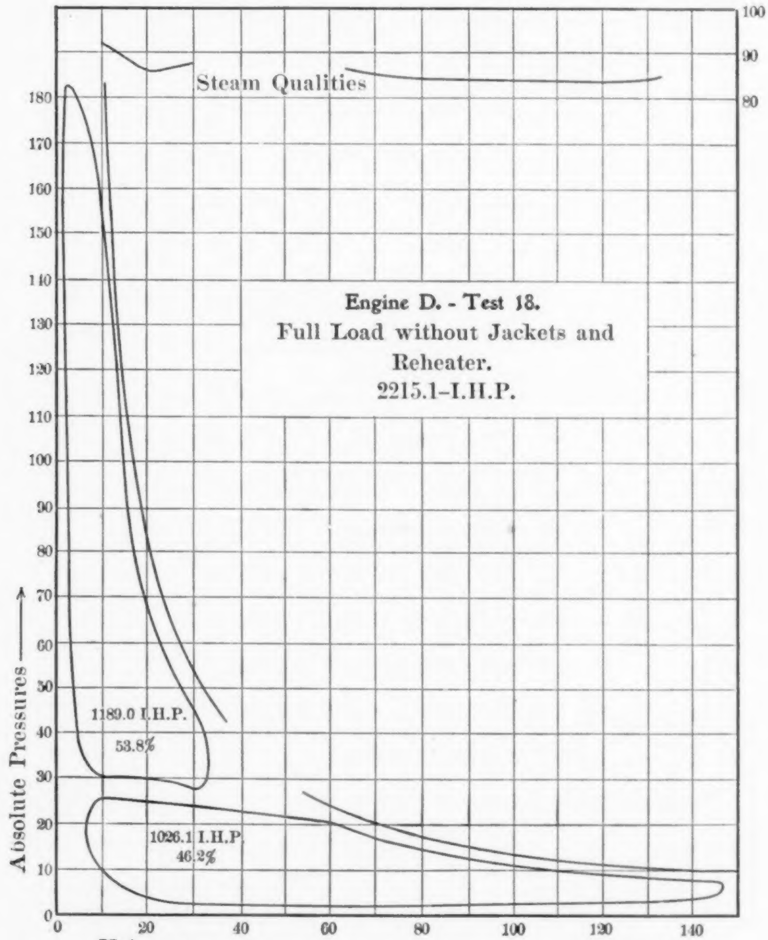


Fig. 139.

ure cylinder that the jacket is effective in this comparatively small cylinder even with moderate superheat at the throttle. The performance of engine H at 20 per cent. overload is considerably better than that of engine B under approximately similar conditions. That the superheat of the admission steam is

probably responsible for this is suggested by the higher quality of steam in the high pressure cylinder. This particular engine, *H*, was known to be in better condition than *G*, its

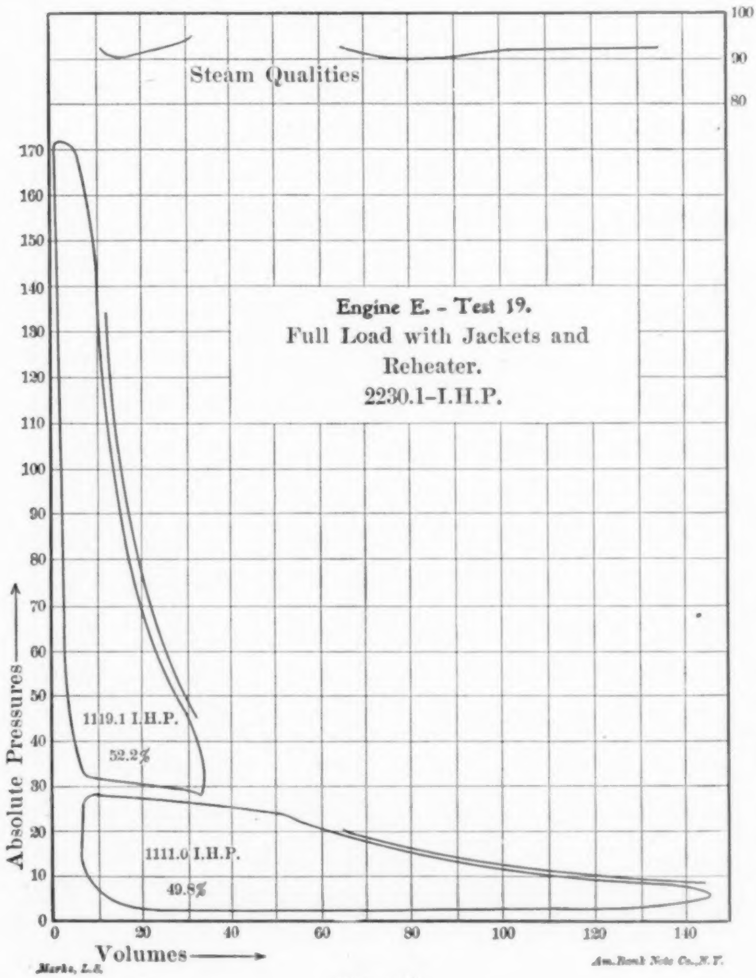


FIG. 140.

pistons were tighter and it had run with unusual smoothness from the beginning. The distinctly better performance of this engine, as compared with the similar engine *G*, is to be accounted for only by the superior condition of the engine.

36. The tests on *K* again emphasize the fact brought out on

tests of C and E that the efficiency throughout the ordinary range of loads is practically constant in these engines, though it naturally falls off when the load is reduced to one-quarter. The

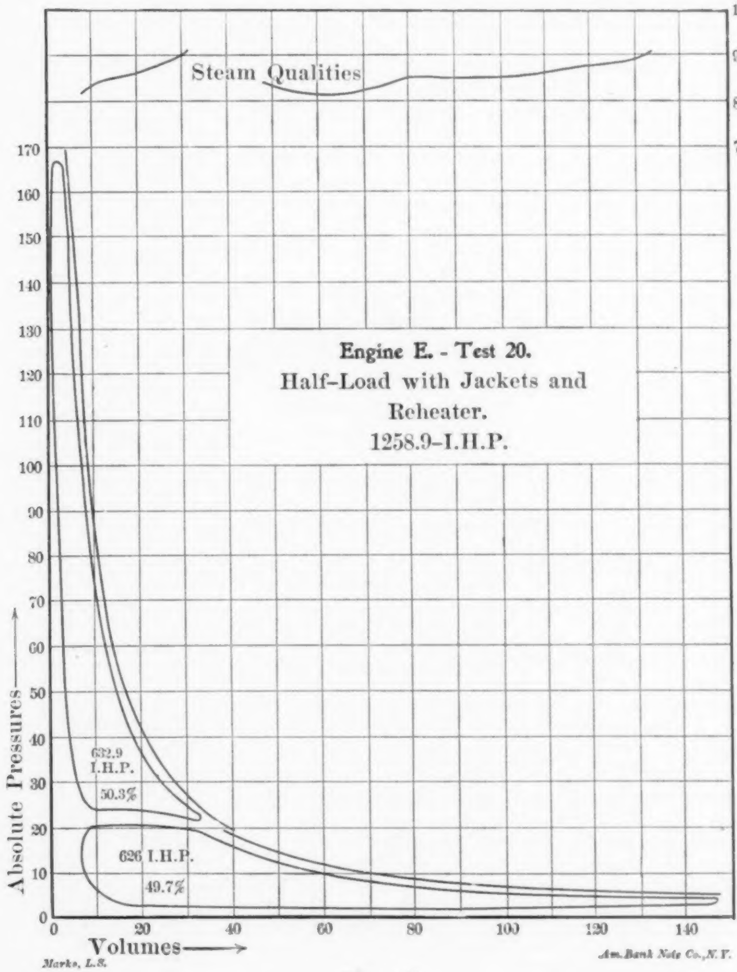


FIG. 141.

advantage of jacketing and reheating is again found to be about 7 per cent.

Temperature-entropy Diagrams.

37. Temperature-entropy diagrams for the tests on engines G, H and K are given in Figs. 150 to 155. There is one omission,

that of test 27. The combined indicator card, Fig. 148, for this test shows that considerable steam returns from the receiver to the high pressure cylinder after release, and the assumption of

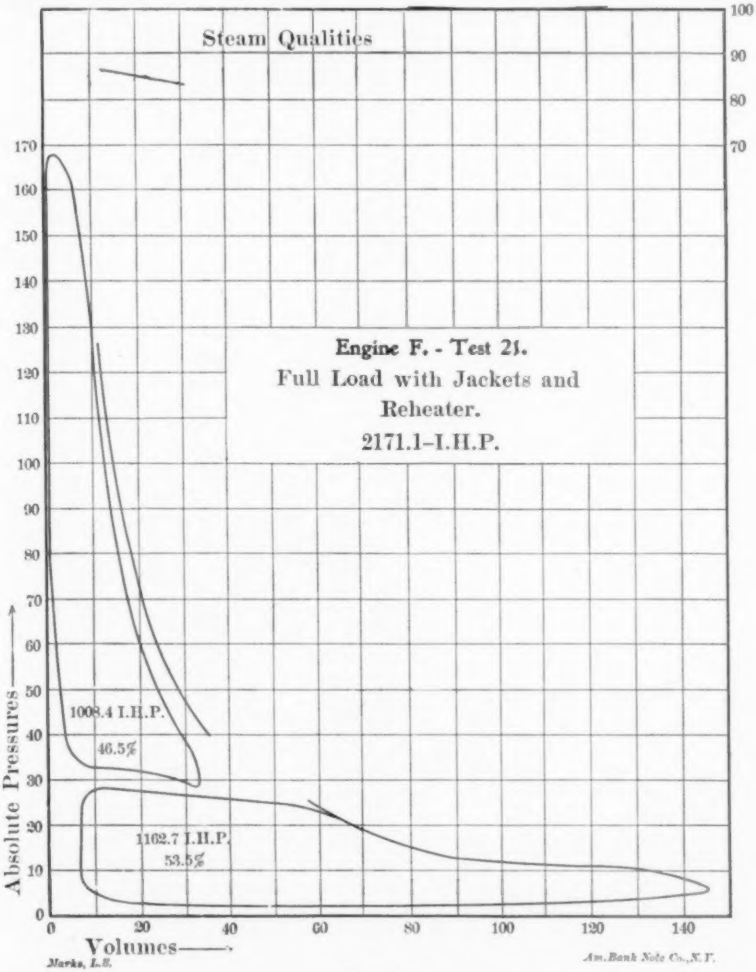


FIG. 142.

constant weight of acting substance, necessary for the construction of the temperature-entropy diagram, would give superheated steam in the exhaust from the high pressure cylinder. It is then impossible to draw a temperature-entropy diagram without grossly distorting the facts.

38. In each of the Figs. 150 to 155 the high pressure and low pressure cards have a common pair of reference curves. As drawn they are valuable mainly in indicating when the heat losses occur. They assume that the total steam acting in each

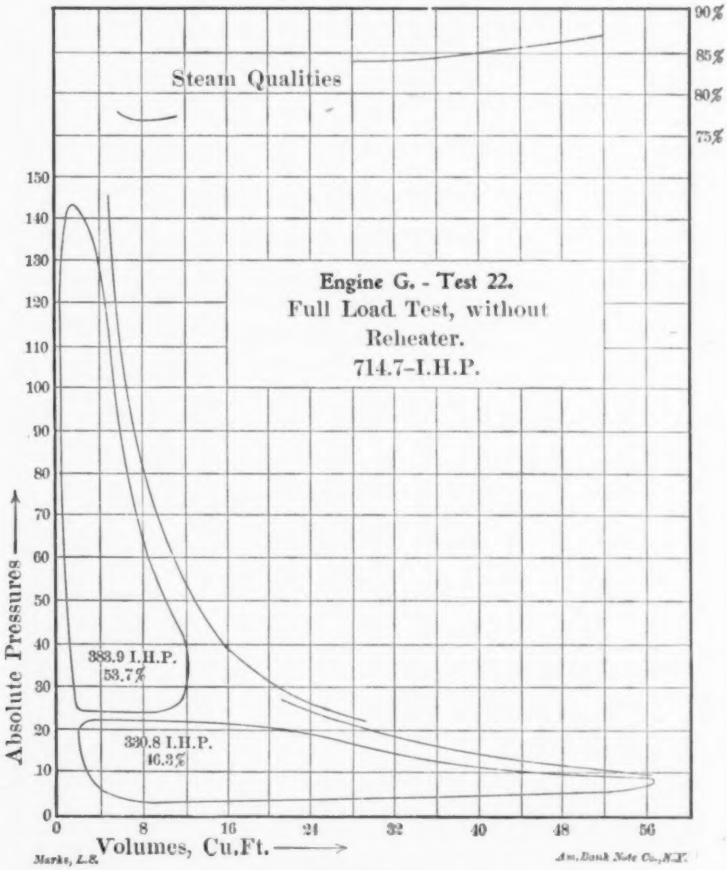


FIG. 143.

cylinder (the sum of the steam admitted and of the clearance steam) weighs just one pound. The relative amounts of work done in the two cylinders will consequently be shown correctly in such a temperature-entropy diagram only in the case when the actual total weights of steam acting during expansion are the same in both cylinders.

39. In the engines under consideration the weights of steam

acting in the two cylinders were different in every case; (*a*) because the weights of clearance steam remaining over in the two cylinders is different in every case, and (*b*) because in many of the tests part of the steam admitted to the high pressure

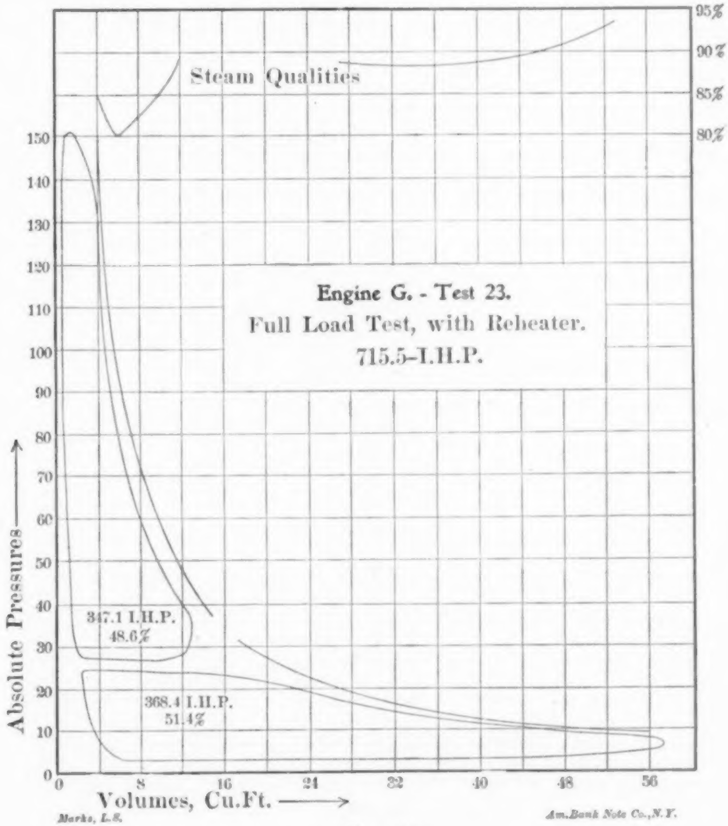


FIG. 144.

cylinder does not go to the low pressure cylinder, but is drained from the receiver. In order, then, to get a correct relation of the temperature-entropy diagrams of the two cylinders, each diagram should be drawn with respect to its own pair of reference curves, and each pair of reference curves should preferably be drawn for the actual weight of steam present during expansion per revolution.

40. Fig. 156 shows the temperature-entropy diagram for the

full load test (No. 25) on engine *K*, drawn to give correct quantitative results for one revolution. The curves *ab*, *cd*, are the reference curves (*i.e.*, the entropy curves for water and for dry and saturated steam) for the high pressure card; the low pressure card has *ef* and *gh* for its reference curves. The position

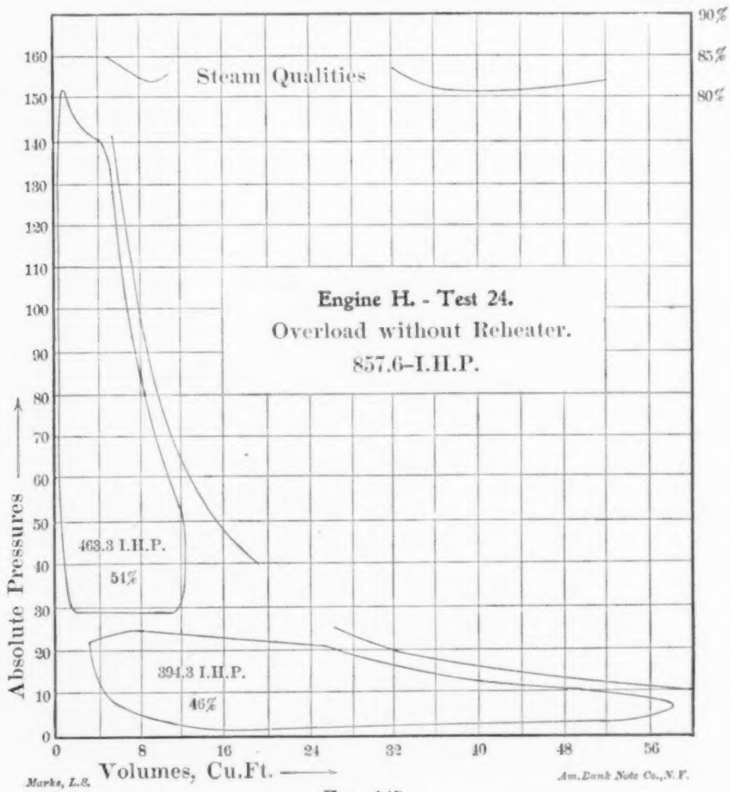


FIG. 145.

of any point on these cards with respect to the proper pair of reference curves gives the quality of the steam at that point on the assumption that all the admission and clearance steam are present in the cylinder. The areas of the cards, when multiplied by the product of the scales of ordinates and abscissæ, give the heat equivalent of the actual work done per revolution.

41. To find the thermodynamic efficiency of each of the cylinders and of the whole engine some additional construction is necessary. The diagram must show the heat added; that is,

the heat of formation of the admission steam measured above its proper starting temperature. It is necessary, then, to eliminate the clearance steam from the diagram. To accomplish this, take the points k and l on the two cards corresponding to the beginnings of compression. At this point of the cycle experi-

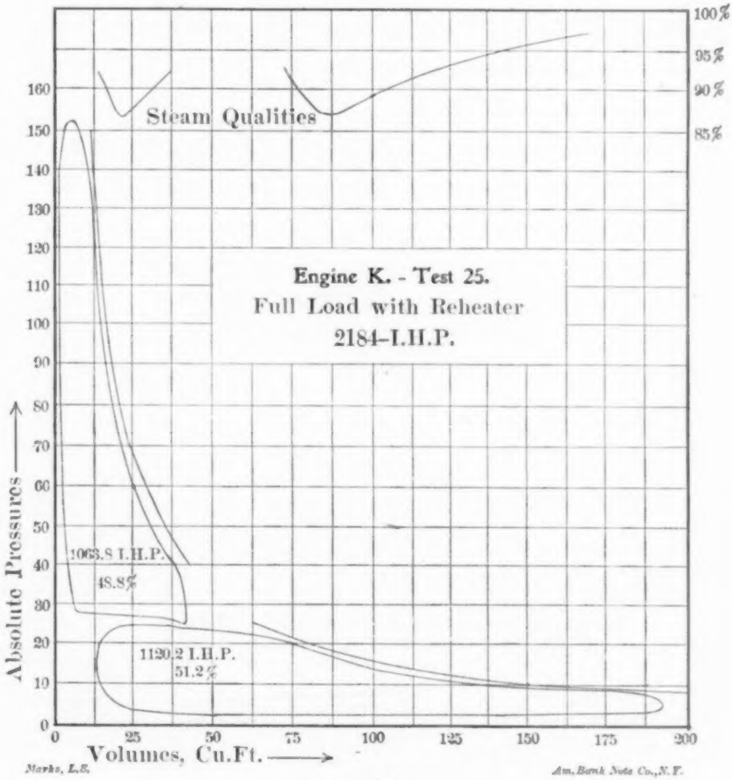


FIG. 146.

ment has indicated, and it is customary to assume, that the clearance steam, which alone remains in the cylinder, is dry and saturated. The point k may be taken as a point on a new reference line giving the entropy of the admission substance as water at that temperature and the new reference curve ko may be drawn for the weight of substance admitted to the high pressure cylinder per revolution. The dry and saturated steam reference curve np for the same weight of substance is constructed as usual by taking horizontal ordinates such as op, kn, of length equal to the

increase in entropy in converting the weight of water substance admitted per revolution from water at a certain temperature into dry and saturated steam at the same temperature. The point *n* of the new reference curve *np* will fall on the curve *cd* for the total steam acting, because the point *k* has its entropy equal to

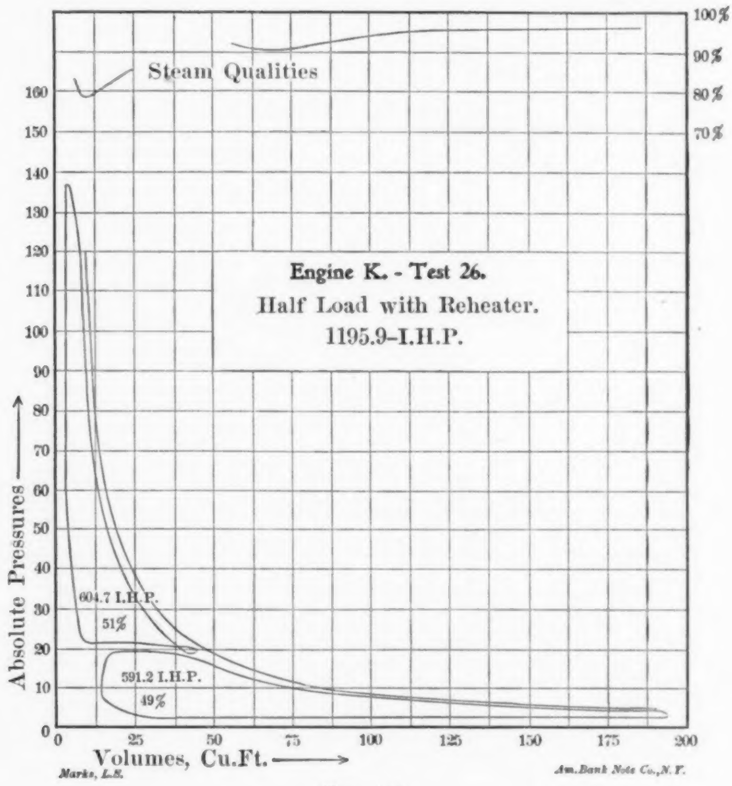


FIG. 147.

the sum of the entropy of the dry and saturated clearance steam plus the entropy of the admission steam as water at the same temperature.

42. In a similar way a new pair of reference curves, *lq*, *rs*, can be drawn for the low pressure cylinder, eliminating the clearance steam.

The area under *Bop* (measured down to absolute zero of temperature) gives the heat supplied to the high pressure cylinder if the steam admitted is dry and saturated; as it is actually

superheated, the area under *Bopt* is the correct measure of the total heat going to the cylinder per revolution. In finding the efficiency of the engine there must be added to this the heat given up in the high pressure jackets and the reheater. The heat given up in these two places cannot be separated in these tests, so it

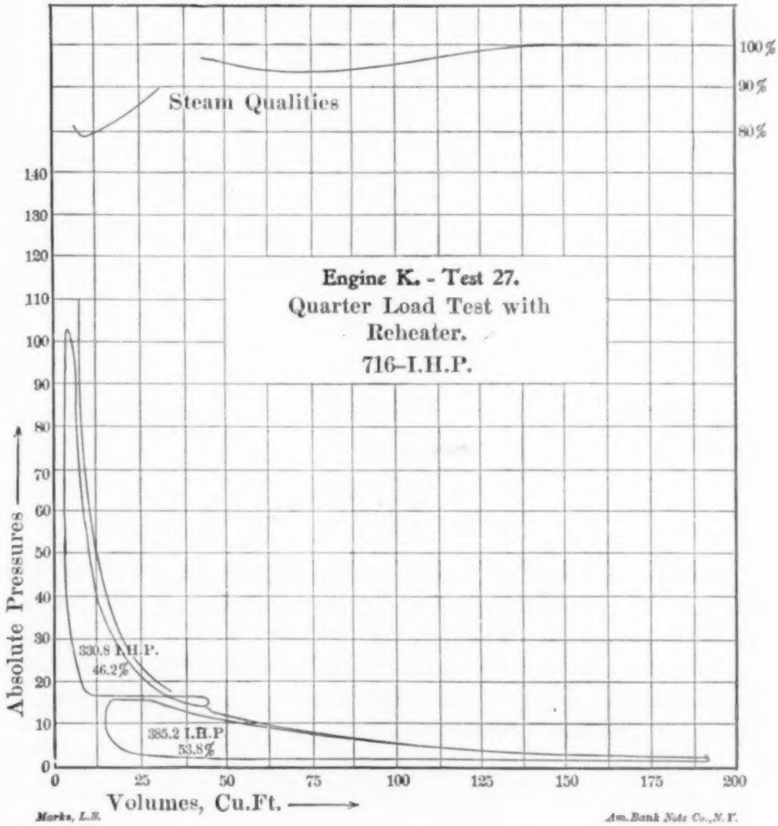


FIG. 148.

will all be assumed to be given to the high pressure cylinder. The steam going to the jackets and reheater gives up only its latent heat and superheat; consequently, if *pu* is the increase of entropy during vaporization of the steam used in the jackets and reheater per revolution, and if *uv* is the entropy line for the superheating of all the steam going to the engine per revolution, the area under *Bouv* gives the total heat going to the engine per revo-

lution. The heat in that part of the steam going through the cylinders is measured above the low pressure exhaust temperature, and that of the steam used in the jackets is measured above the saturation temperature of steam at the admission pressure. The area under *Bouv* gives then the total heat supply per revolution measured above the ideal feed temperature. The work done by the ideal (Rankine) engine working under the same conditions is represented by the area *BouvD*. The efficiency of the actual engine compared with the ideal

$$= \frac{\text{H. P. area} + \text{L. P. area}}{B\,o\,u\,v\,D}.$$

The efficiency of the high pressure cylinder, considered as a separate engine, compared with the ideal

$$= \frac{\text{H. P. area}}{x\,o\,u\,v\,w}.$$

The efficiency of the low pressure cylinder compared with the ideal

$$= \frac{\text{L. P. area}}{A\,q\,s\,y\,C}.$$

43. It is to be noted that Rankine's unjacketed cycle has been used as the ideal although the high pressure cylinder is jacketed. It is also to be observed that the efficiency of the high pressure cylinder as defined above is less than its actual efficiency because of the assumption that the heat of the reheater steam goes to the high pressure cylinder.

The Value of High-pressure Jackets and of Reheating.

44. There are certain facts which the writer believes may be postulated with reference to the effectiveness of jacketing and reheating, and which are supported by the results of these tests.

45. The saving by jacketing varies with the following factors:

(*a*) It increases as the cut-off becomes earlier.

(*b*) It decreases as the superheat of the entering steam increases.

(*c*) It decreases with increase in size of the engine.

46. There is no saving by reheating when the reheater does

no more than to dry the steam. If there is any advantage in using dry and saturated steam in the low pressure cylinder over the use of wet steam, it can be obtained by the use of a separator between the cylinders. In tests 2, 7, 9 and 24, when the reheater was not in use, the receiver acted as an almost perfect

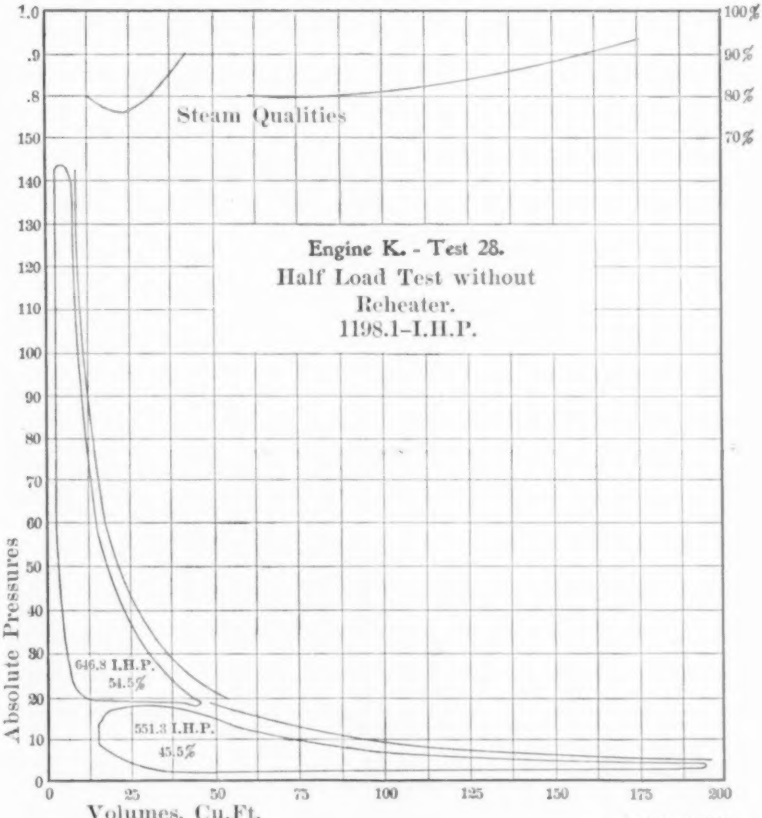


FIG. 149.

separator, taking away all the condensation which calculations showed to have occurred in the high pressure cylinder. If the reheater merely vaporizes the condensed steam at the expense of a practically equal quantity of high pressure steam it is not only non-effective but is also, probably, a source of actual loss, since more work could have been obtained from the total steam used if it had all gone into the high pressure cylinder. The reheater

then should be regarded merely as a superheating device for the steam entering the low pressure cylinder, and it may be expected to be more effective the greater the amount of superheat it gives the receiver steam. It is probable that the reheater would be more effective if the only work were superheating, as it might be if the wet steam exhausting from the high pressure cylinder passed through a good separator before reaching the reheater.

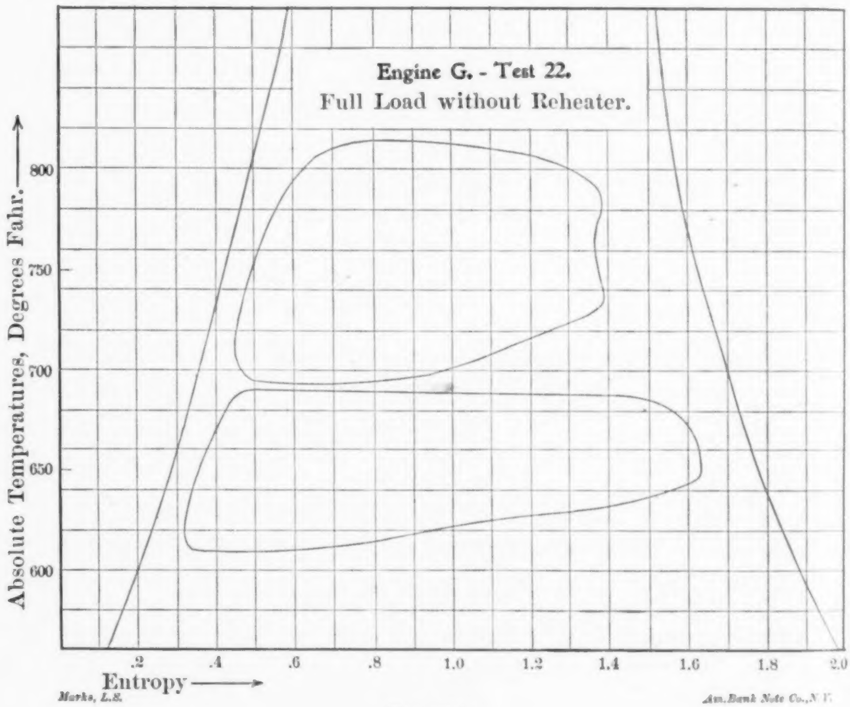


Fig. 150.

As the reheater will have less to do when the engine is running at low loads it may be expected to give a higher superheat at low loads, and consequently to be more effective. With the above in mind, an examination of the tests already described (the principal results of which are collected in Table V. for more easy inspection) throws some light on the conditions under which the jacketing of the high pressure cylinder and the practice of reheating are desirable, and shows the saving to be expected from them in large size engines.

TABLE V.

Principal Results of All the Tests.

1	Number of test	1	2	3	4	5	6	7	8	9	10	11	12	13	14
2	Engine	A	A	B	B	B	B	B	B	B	B	B	B	C	C
3	Nominal load	1	1	¼	⅛	½	¾	¾	1	1	1¼	1¼	1	1	1
4	With or without jackets and reheater	With	Without	With	With	Without	With	Without	With	Without	With	Without	With	With	Without
5	Nominal H. P. cylinder diameter, inches	28	28	23	23	23	23	23	23	23	23	23	23	29	29
6	Nominal L. P. cylinder diameter, inches	58	58	48	48	48	48	48	48	48	48	48	48	60	60
7	Stroke, inches	48	48	48	48	48	48	48	48	48	48	48	48	56	56
8	Steam pressure at throttle, pounds per square inch	159	161	152.6	151.5	155.3	150	155.1	149.5	155.9	152.6	155.2	161.8	168.6	167
9	Superheat at throttle, Fahr.	Wet	Wet	.8	4.8	21.8	10.7	17.5	9.2	14.6	8	13.8	18.6	80.2	78.5
10	Superheat entering L. P. cylinder, Fahr.	35.5	Wet	99.8	75.4	Wet	60.5	Wet	46.6	Wet	26.5	Wet	49.4	60.3	Wet
11	Vacuum in exhaust pipe, inches	25	24.5	28.2	28.0	27.25	28.3	27.1	27.7	28.0	25.2	26.8	27.0	24.9	24.7
12	I. H. P.	1,994	1,984	309	552	548	845	831	1,047	1,105	1,498	1,516	1,104	2,267	2,240
13	Total steam per I. H. P. per hour	13.59	13.68	14.04	13.71	15.11	13.63	14.72	13.59	14.41	14.57	15.16	13.58	12.72	12.98
14	Per cent. of total steam used in jackets and reheater	6.8	......	10.2	8.95	......	7.91	......	7.49	......	8.9	......	6.9	6.5	......
15	Per cent. of cylinder steam drained from receiver	1.6	6.0	......	......	3.7	......	6.7	......	5.97	......	4.97	......	......	2.3
16	Quality of steam at cut-off H. P. cylinder	81.6	75.5	61	58	60	68	72	68	60	79	74	65	86	86
17	Quality of steam at release H. P. cylinder	85.6	79.6	85	85	88	80	74	78	75	83	78	81	83	88
18	Quality of steam at cut-off L. P. cylinder	92.4	87	95	90	65	88	78	90	76	94	88	90	99	84
19	Quality of steam at release L. P. cylinder	95.5	93	95	95	91	88	90	91	88	98	89	93	96	91
20	B. T. U. per I. H. P. per minute	245	248	258	252	277	251	270	248	267	255	265	241	236	244
21	Thermodynamic efficiency, per cent.	17.3	17.1	16.5	16.9	15.4	17.0	15.8	17.2	15.9	16.7	16.1	17.6	18.0	17.4
22	Thermodynamic efficiency compared with ideal cycle	65.4	65	58	60	58	60.5	59	62.5	58	66.1	61.5	64.5	68.3	65.6
23	Per cent. saving by jacketing and reheating	1.3	......	......	9.0	......	8.0	......	7.0	......	4.0	......	......	3.0	......
24	Engine friction at no load, I. H. P.	118	......	......	......	......	......	......	70.5	......	......	......	......	159	......
25	Mechanical efficiency at rated load, per cent.	94.1	......	......	......	......	......	......	94	......	......	......	......	93.2	......

1	Number of test	15	16	17	18	19	20	21	22	23	24	25	26	27	28
2	Engine	C	C	D	D	E	E	F	G	G	H	K	K	K	K
3	Nominal load	¾	⅛	1	1	1	½	1	1	1	1¼	1	½	¼	½
4	With or without jackets or reheater	With	With	With	Without	With	With	With	Without	With	Without	With	With	With	Without
5	Nominal H. P. cylinder diameter, inches	29	29	29	29	29	29	29	18	18	18	31	31	31	31
6	Nominal L. P. cylinder diameter, inches	60	60	60	60	60	66	60	38	38	38	64	64	64	64
7	Stroke, inches	56	56	56	56	56	56	56	42	42	42	48	48	48	48
8	Steam pressure at throttle, pounds per square inch	167	169.4	162.8	163.1	159.2	160.4	158.7	186	137	131	136	132	136	135
9	Superheat at throttle, Fahr.	64.5	48.3	98.4	98.4	72.4	70.4	92.7	72.2	75	55.5	64.2	70.3	55.3	73.5
10	Superheat entering L. P. cylinder, Fahr.	67	84.8	60.5	Wet	60.9	86.0	61.7	Wet	49.1	Wet	40.3	59	79.1	Wet
11	Vacuum in exhaust pipe, inches	25.6	26.1	25.6	25.7	25.3	25.4	26.1	26.2	26.3	26.1	25.4	26.4	26.4	25.8
12	I. H. P.	1,831	1,239	2,202	2,215	2,230	1,259	2,171	715	716	858	2,184	1,196	716	1,198
13	Total steam per I. H. P. per hour	12.71	12.38	11.57	11.88	11.67	11.78	11.67	14.13	13.21	13.13	12.50	12.22	12.95	12.90
14	Per cent of total steam used in jackets and reheater	7.2	9.3	7.4	......	7.1	9.3	5.6	......	6.45	......	4.89	7.28	9.05	......
15	Per cent. of cylinder steam drained from receiver	......	......	......	......	......	......	......	3.35	.69	6.12	.93	.67	......	2.75
16	Quality of steam at cut-off H. P. cylinder	77	65	99	93	93	82	89	76	85	86	92	83	81	80.5
17	Quality of steam at release H. P. cylinder	82	80	94	87	94	92	84	75	88.5	88	93	82.5	88	92
18	Quality of steam at cut-off L. P. cylinder	93	99	100	86	93	85	100	84	89	84	93	92	97.4	80.5
19	Quality of steam at release L. P. cylinder	98.5	99	96	84	93	92	100	87	93.2	82	97	96	100	94
20	B. T. U. per I. H. P. per minute	235	227	217	226.5	217	217	221	264.9	245.5	243.3	227.3	226.6	241.5	244
21	Thermodynamic efficiency, per cent.	18.1	18.7	19.6	18.7	19.6	19.6	19.2	16.0	17.3	17.4	18.7	18.7	17.6	17.4
22	Thermodynamic efficiency compared with ideal cycle	67.3	67.7	75.4	72.3	72.6	73	70.7	62.8	67.4	68.5	74.2	73.3	66.8	67.5
23	Per cent. saving by jacketing and reheating	......	......	4.5	......	......	......	......	......	7.5	......	......	7.2	......	......
24	Engine friction at no load, I. H. P.	......	......	169	......	164	......	197	......	50	50	......	......	......	......
25	Mechanical efficiency at rated load, per cent.	......	......	92.7	......	93	......	91.4	......	93.4	93.4	......	......	......	......

47. In engine A, where saturated steam is used, the jackets, at full load, keep the steam about 6 per cent. dryer in the high pressure cylinder and the reheater superheats 35 degrees. As the greater part of the 6.8 per cent. of the total steam which is condensed in the jackets and reheater must have been condensed in the latter place, it is obvious that the reheater cannot have contributed anything to the small total saving of 1.3 per cent.

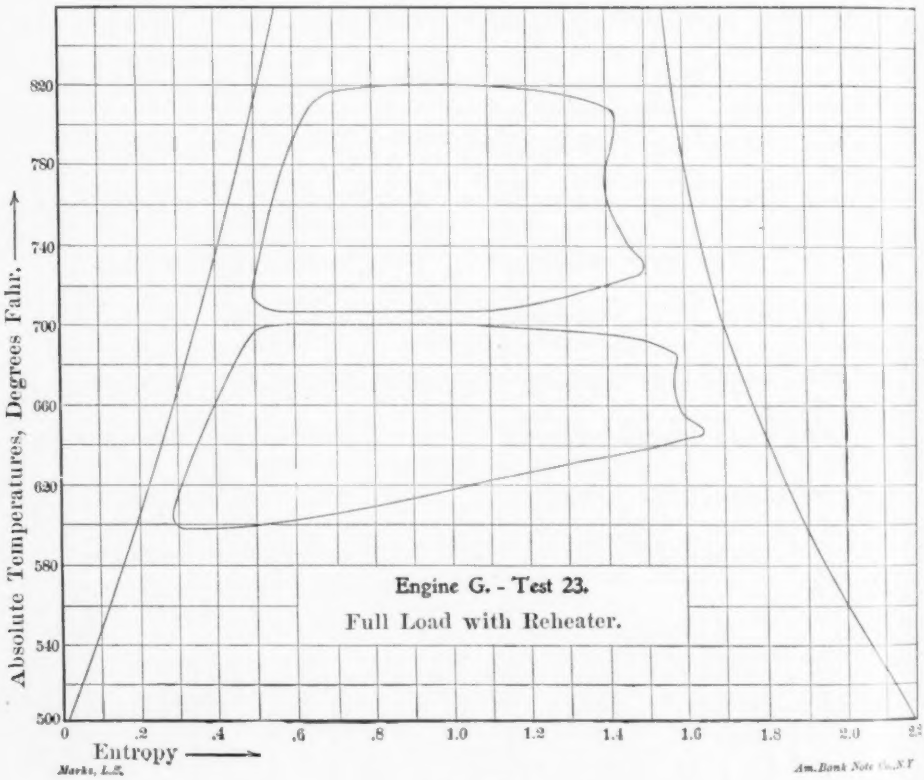


FIG. 151.

48. The conditions in engine B are much more satisfactory for effective action of jackets and reheater, especially at the lower loads. The engine is smaller, and its reheater is more effective. At half load, with 75 degrees superheat of the receiver steam, the saving was 9 per cent.; at three-fourths load with 60 degrees superheat there was 7 per cent. saving; at full load with 46 degrees superheat there was still 7 per cent. saving, and even at one-

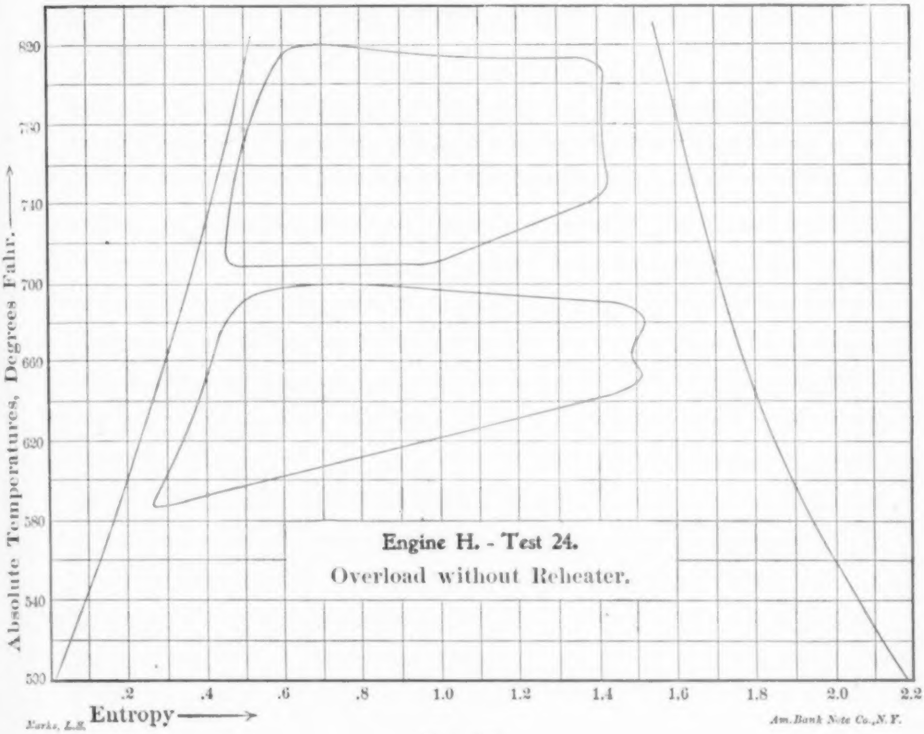


FIG. 152.

quarter overload with 26 degrees superheat there was 4 per cent. saving.

49. The larger engines *C* and *D* with 80 degrees and 98 degrees initial superheat and 60 degrees superheat by the reheater show but 3 per cent. and 4.5 per cent. saving respectively.

50. The engine *G* is only one-third the power of engines *C* and *D*, consequently the jackets are much more effective (raising the steam quality 10 per cent. in the high pressure cylinder), so that with 49 degrees superheat going to the low pressure cylinder the saving is 7.5 per cent.

51. The tests on engine *K* at half load, with 59 degrees superheat by the reheater, show 7.2 per cent. saving. The engines *C*, *D* and *K* have sufficient initial superheat and are of such size as to make the high pressure jackets of but little value, so that the savings shown are due principally to the action of the reheater.

52. A study of the above results appears to indicate that the reheater will not justify its use (except as a separator) unless it superheats the low pressure admission steam at least 30 degrees. An examination of the qualities at release in the low pressure cylinders indicates that 100 degrees superheat of the receiver steam will probably be enough to make the steam dry and satur-

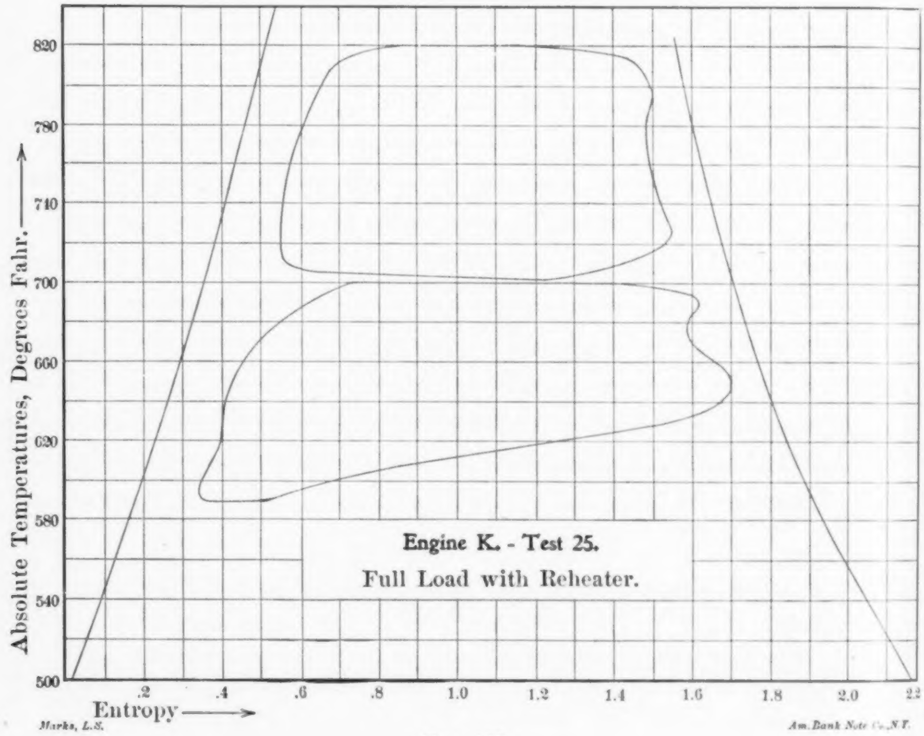


FIG. 153.

ated at release. As it is not desirable to have superheated steam at release this suggests the probable desirable limit to the amount of superheat to be given by the reheater.

The Value of Moderate Superheating.

53. The engines *C, D, E, F, G, H* and *K* were all supplied with steam from Babcock and Wilcox boilers, fitted with superheaters giving from 100 degrees to 125 degrees superheat at the boiler when running at the rated power. The amount of super-

heat at the engine depends on the load at which the engine is running; (*a*) because the superheat at the boiler decreases as its load is decreased, and (*b*) because the fall of temperature in the steam pipe increases as the weight of steam passing through it diminishes. For these two reasons the superheat was less at low loads than at higher loads except in some cases where the number

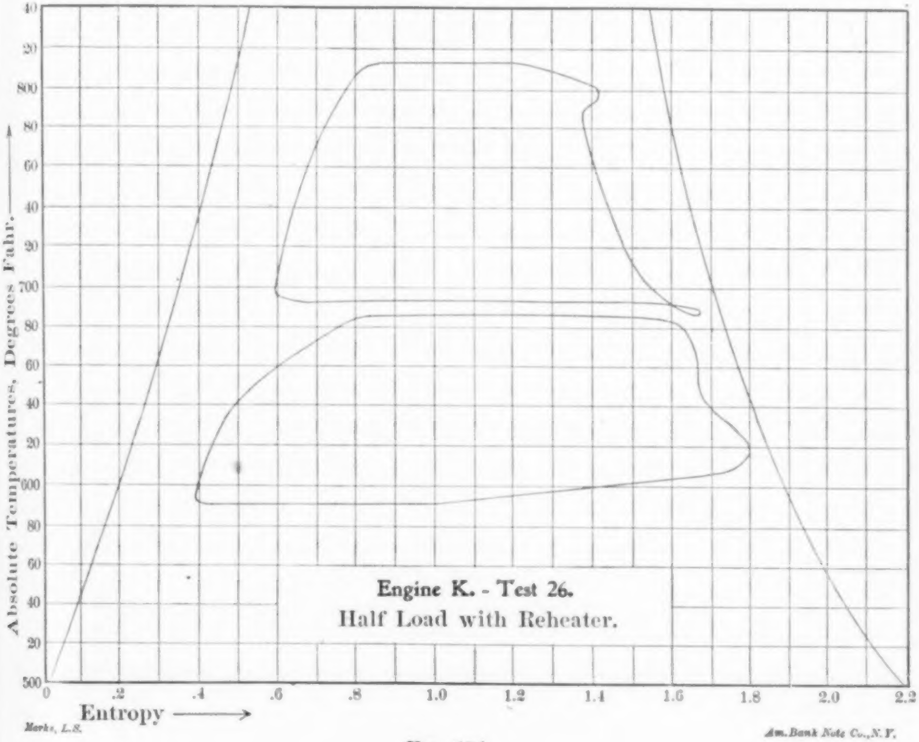


FIG. 154.

of boilers used could be decreased as the load decreased. As the superheat going to the low pressure cylinder when the reheaters were in use varied in the opposite way; that is, increased with decrease of load, these two variations tended to offset one another in their influence on the engine efficiency. The tests without the reheaters in use will then be the most valuable for showing the influence of the superheat of the high pressure steam. The tests at full load show that engine *A* uses 248 British thermal units per indicated horse-power per minute with

no superheat; engine C of about the same size with 78 degrees superheat uses 239 British thermal units, and engine D with 98 degrees superheat uses 226 British thermal units, a saving of about 9 per cent.; some of which, however, is also due to a better vacuum and better cylinder proportions. Engine B, a smaller engine, uses 267 British thermal units with 15 degrees super-

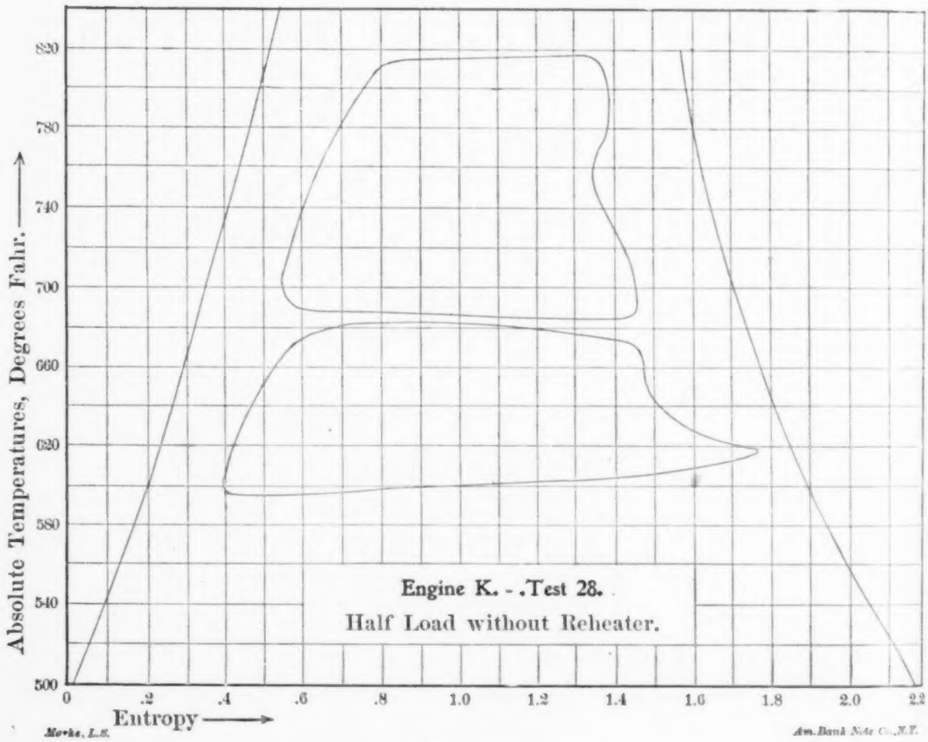


FIG. 155.

heat—which is practically the same result as that obtained from G, a still smaller engine, with a poorer vacuum but with 72 degrees superheat. In the test of D with 98 degrees superheat the quality of the steam at release is 87 per cent., so that it is evident that when the jackets are not used a much greater superheat is desirable in order to prevent condensation in the high pressure cylinder—probably at least 150 degrees will be necessary. An even greater superheat will be necessary to keep the steam dry in both cylinders. With the jackets in use and with

98 degrees superheat the quality in the high pressure cylinder at cut-off is 99 per cent., and at release is 94 per cent. The advantage gained by superheating is, of course, greater in the smaller engines.

The Variation of Economy with Engine Load.

54. The engines *B*, *C*, *E* and *K* were all tested at several loads so as to determine the effect of variation of engine load on the

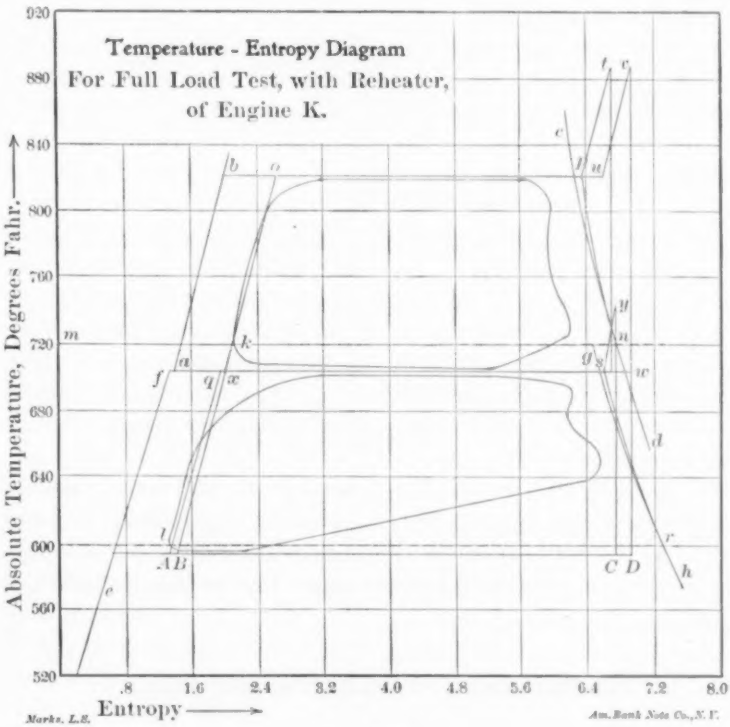


FIG. 156.

efficiency of the engine. In all the cases (engines *C*, *E* and *K*) where moderately superheated steam was admitted to both cylinders the important fact was brought out that the heat consumption per indicated horse-power is practically constant through a range of load varying from one-half load to full load, and probably even to a considerable overload. The apparent exception in the better performance of engine *C* at half load is

probably due to a better vacuum at that load. The general result was to be expected because the effect of superheat is to reduce the amount of heat disappearing during admission, and, consequently, to permit the increased expansion at low loads to occur without excessive cylinder condensation. In engine *B*, with very small superheat of the admission steam, there is a slight decrease in economy as the load decreases from full load, and in the tests without reheater, and consequently with no superheat going to the low pressure cylinder this decrease in economy is very marked.

55. It is, perhaps, hardly necessary to emphasize the fact that the constancy of heat consumption referred to above is in terms of the indicated horse-power. In all these engines the friction horse-power is low, and the mechanical efficiencies at the rated loads are high, varying (see Table V., line 25) from 91.4 per cent. to 94 per cent. and averaging 93.2 per cent. The friction horse-powers were determined by taking cards with no load on the engine, but they really represent more than the friction of the engine proper, since they include some losses properly chargeable to the generator—such as the brush friction, the armature windage, the bearing friction of the armature and, in some cases, a low excitation of the field. Consequently, the real mechanical efficiencies of the engines are somewhat higher than the quantities given in the table.

56. The low friction of the engine causes the heat consumption per electrical horse-power to change a comparatively small amount as the load decreases; the heat consumption per kilowatt per hour will be some 6 or 7 per cent. greater at half load than at full load.

The Radiation and Conduction Heat Losses.

57. In those tests where the reheater was in use, the knowledge of the exact condition of the steam entering the low pressure cylinder permitted the determination of the heat lost by radiation and conduction from the high pressure cylinder and the receiver. The total heat of the steam coming to the engine is the sum of the four following quantities :

(1) The heat going to the low pressure cylinder.

(2) The heat equivalent of the work done in the high pressure cylinder.

(3) The heat escaping with the reheater and receiver drainage.

(4) The heat lost by radiation and conduction from the high pressure cylinder and the receiver.

The first three quantities can be calculated from the observations made on the test, and consequently, the last quantity can be determined by the heat balance stated above. The radiation and conduction loss was calculated for most of the tests, and was found to vary from $\frac{1}{2}$ to 1 per cent. of the total heat supply to the engine at full load. As the low pressure cylinder is as carefully lagged as the high pressure cylinder it appears that from 1 to $1\frac{1}{2}$ per cent. of the total heat supply to the engine at full load will be lost by external radiation and conduction. The larger percentage applies to the smaller engines.

Piston Leakage.

58. The steam qualities at cut-off and release, given in lines 16 to 19 of Table V., show considerable variation in different engines running under practically the same conditions. The reason for this variation is apparently not far to seek, and depends on a phenomenon to which but small consideration is generally given in the discussion of steam engine performances. This phenomenon is piston leakage.

59. Most of the engines were tested when at rest for piston leakage, before the runs were made, and in no case was there any but slight leakage. It is probable, however, that a piston which is quite tight when at rest will leak when running. The static leakage tests were made for a small number of piston positions and did not insure static steam tightness in every position. There is evidence moreover to show that even if the piston is tight in every position when at rest, it may leak when in motion owing to the breaking up of the oil film on the cylinder walls. If to the results of the leakage test of the piston is added the knowledge that an intelligent engineer has of the condition of the cylinders of which he has had charge, it is probable that a more accurate statement can be made as to the tightness of the pistons. From such data the following statements may be made as to the condition of the engines tested.

Engine *B*, neither piston perfectly tight, but both in good condition.

Engine *C*, no appreciable leakage.

Engine *D*, both cylinders in very good condition.

Engine *E*, high pressure cylinder very good; low pressure piston had not worn down to maximum tightness.

Engine *F*, high pressure cylinder had been scored a few weeks before test and had not worn quite tight; low pressure cylinder unusually good.

Engine *H*, high pressure cylinder very good; both cylinders better than engine *G*.

60. The above conditions, as known before the tests, will be found to explain most of the variations in steam quality to which reference has been made. For example, of the two similar engines *E* and *F*, the latter shows lower quality at cut-off in the high pressure cylinder, notwithstanding a greater initial superheat—and this quality is seen to decrease throughout expansion. Leakage past the high pressure piston readily accounts for this. In the low pressure cylinders of these two engines, the phenomenon is reversed, and the remarkably high quality of the steam in engine *F* is presumably due to the unusually good condition of the cylinder.

61. Similarly comparing tests 22 and 24 on the exactly similar engines *G* and *H* under practically similar conditions, a marked advantage is seen in the quality of the steam during expansion in the high pressure cylinder during the latter test—a result to be expected from the known better conditions of that cylinder.

62. These examples could be multiplied were it desirable. The effect of the piston leakages, when moderate, on the engine economy is not very great since steam leaking by the high pressure piston will be available for doing work in the low pressure cylinder.

Conclusions.

63. In summing up the general results of the tests the following conclusions appear to the writer to be justified when applied to large size, high-speed, compound, four-valve engines of common proportions.

The jacketing of the high pressure cylinder is of but little value when moderately superheated steam (100 degrees Fahr.) is used.

Reheating is probably a source of loss unless it superheats the receiver steam at least 30 degrees Fahr., and is not fully effective unless it superheats about 100 degrees Fahr. In the latter case

it may be expected to effect a saving of 6 to 8 per cent. of the total heat used per indicated horse-power.

Jacketing the low pressure cylinder is shown by the steam qualities during expansion in the low pressure cylinder to be unnecessary and therefore undesirable when the reheating is effective. The effect of admitting moderately superheated steam to both the high pressure and low pressure cylinders is to keep the heat consumption per indicated horse-power practically constant throughout a considerable range of loads—from half load to about one-quarter overload.

The variation within the ordinary limits of the ratio of stroke to diameter in large size engines of the same power when using moderately superheated steam, does not have any marked effect upon the economy of the engine. The size of the engine is an important factor in determining its efficiency. The engine *G* has about 10 per cent. greater heat consumption per indicated horse-power than *K*, which is three times larger.

DISCUSSION.

Mr. W. R. A. Harris.—As the discussion on above was closed rather early, I should like to pass a few remarks on the subject.

First. I think the tests to have been satisfactory should have been all of one duration, and should have gone farther than steam economy, which, for superheating, is not sufficient, coal consumption being the essential point; for if the steam economy gains 10 per cent. at a cost of 15 per cent. more fuel owing to superheating there is a loss.

Second. It would be well to know the type of superheater and distance from engine, *i. e.*, length of pipe, etc.—as it is a small amount of superheat.

There are many places on the Continent and in England where superheat is used as high as 650 degrees Fahr. to 700 degrees Fahr., with an economy in *coal* consumption of 7½ per cent.

Another point to bear in mind in favor of superheating, for new plants, is the increased velocity of superheated steam over saturated, enabling smaller pipes to be used and a greater piston speed maintained.

Mr. George Barrus.—I have not had an opportunity to study this paper very carefully, but I have noticed one or two omissions in it. For example, the author states

that the tests on engine *A* were conducted by himself. Also the tests on engines *G*, *H* and *K;* but tests *B*, *C*, *D*, *E* and *F* were conducted by Mr. Parker and Mr. Cook, members of the Society, and by two other parties. It would be interesting if he stated why the results of the tests made by these gentlemen are incorporated in his paper.

I notice in all the tables what is termed "quality of steam at cut-off and release." I do not think that expresses what the author really means. I suppose he means what is usually called the "percentage of steam accounted for by the indicator." This percentage shows how much steam appears on the indicator card, but it does not show the exact condition of the steam as to the amount of moisture it contains, as would be understood from the use of the term "quality." I think the tables ought to be revised to meet this objection.

Then it seems to me the tables ought to be arranged in accordance with the forms established by our committee on engine testing, and if they are not arranged in that way there should be some explanation about it. If those forms are of no use let us know it. If they are all right the author, as a member of the Society, ought to follow them.

There are no copies of the original indicator diagrams in this paper. The author should supply them for each of the tests. The reason I make this statement is, that I saw the report of the test on engine *A*, as well as some of the original diagrams which were taken. I was struck with the fact that the diagrams were poor. I do not mean that they showed poor engine performance, but they showed poor indicator work. You all know that if a vertical engine is not solid on its foundation and the indicator outfit is not properly connected, the indicator is very unsteady, and the diagrams obtained will likewise be shaky and unreliable. These diagrams were very poor in that respect, and if it had been my test I should have placed no reliance on their indications.

Mr. C. V. Kerr.—An investigation that I had the privilege of making about a year and a half ago on the theory and practice of superheated steam, convinced me that the reheater would come again into solid favor with the general use of superheated steam. When the reheater receiver is used with saturated steam the trouble is that we cannot get a sufficient amount of reheating surface; you cannot put enough reheating surface into the

reheater to secure as much superheat as you will need, and tests have shown this quite conclusively. In the 5,000 horse-power engine at the Waterside station, in New York, which was recently tested, it was shown that both jackets and reheater, either together or alone, were practically useless throughout the working range of load. How much heat can you expect to get into a cylinder full of steam in a fraction of a second through cylinder walls that may be five or six feet in diameter? That difficulty, coupled with the fact that in the reheater with saturated steam you cannot get much superheat, is the reason that tests show both jacket and reheater to be nearly useless. I think that with 200 degrees superheat and a receiver with an adequate amount of superheating surface, it is possible to get enough superheat to keep down condensation to the point, at least, of cut-off, and that will solve the problem of initial condensation.

If you will note the table on page 49, in which the principal results of all the tests are given, and look over the various tests, you will find this to be true, I believe, that with the highest superheat at the throttle and with 60 degrees superheat at the low pressure admission, and perhaps less than the average vacuum, we find the lowest steam consumption in the 28 tests, which is 11.57 pounds per indicated horse-power hour. We find 100 per cent. as the quality of the steam at cut-off in the low pressure cylinder, the lowest heat consumption in British thermal units per minute, and the highest dynamic efficiency compared with the ideal cycle. This seems to me to show that the superheating receiver is bound to come into favor with the general use of superheated steam.

Mr. George I. Rockwood.—The author gives the conclusions he derives from these tests at the end of his paper, which is commendable; but after all they hardly seem to justify so much labor or to be especially valuable or new. On the contrary, the paper seems to be merely a threshing over of old straw.

The performances of all of these engines have been beaten many times by other engines of the same make and type; they were also beaten many years ago by an engine of only 200 horsepower having a higher cylinder ratio. I maintained twelve years ago that compounds having a higher cylinder ratio would beat those with the ordinary ratio by 10 or 15 per cent. To see how that view would apply in the present case, these tests speak of 13.5 pounds of steam per indicated horse-power per hour on an engine

of 2,400 horse-power (which is really a very poor performance). Mr. Barrus tested an engine a year ago of 500 horse-power, in which test the performance was 11.2 pounds. This engine was a high-ratio compound, and beat the 2,400 horse-power engine by 17 per cent., both engines operating "under similar external working conditions." This would seem to call in question the conclusion that the smaller engine is about 10 per cent. less economical than the larger. This conclusion of the author appears to be general and not confined to these engines alone.

I am inclined to agree that the supposition that the larger the engine *of a given type,* the greater its economy will be is true; but hardly think these tests prove it. Even if the test of the 500 horse-power high-ratio compound be compared with that of the 2,500 horse-power high-ratio compound engines at the Waterside station, to which Mr. Kerr has just referred, it will be seen that the smaller engine is still the most economical of all. This, however, comes from the fact that the Waterside engines, being designed for use with superheated steam, had poppet valves, which involved 15 per cent. clearance in the first cylinder, with a resulting total loss of economy of 6 per cent.

With regard to the value of reheaters, the author does not note two or three important considerations. The principal one is that a reheater increases the economical power of a given engine to an extent out of all proportion to its cost. Even, therefore, if it does not increase the economy of the engine—and I feel very sure that if properly designed it cannot decrease the economy— its use is more than justified. Nothing is said, I believe, about draining the water of condensation from the high pressure exhaust pipe before the steam meets the reheating surface. This is necessary if economy is aimed at.

Some people still like to test steam-engines. To me the subject is of only passing interest, as I believe the steam-turbine has routed the reciprocating machine completely for all work which can be done by electrical transmission of the power; and there is now hardly to be found any place where the motor is not of greater advantage than shafting and belting.

*Prof. L. S. Marks.**—The writer regrets that Mr. Barrus did not have an opportunity to study this paper very carefully before he criticised it. He will find his criticism in part answered

---

* Author's closure, under the Rules.

in the paper itself, but as the criticism includes also other points, the writer will endeavor to reply fully.

In paragraph one is given the reason why the writer included with his own tests all the available unpublished tests on engines of the same make. It may also be pointed out that these other tests were not published by those employees of the company who had conducted the tests, because of a special regulation of the company forbidding such action on their part; and that their inclusion in this paper was with the express permission of the president of the company.

The term "quality of steam at cut-off" was used because of its greater brevity, and in the belief that the members of this Society understand what is meant hereby; and further, the meaning of the term is made abundantly clear by the discussion under the heading of "piston leakage." The tests were presented with a special object in view, and the writer, while appreciating fully the excellent work done by the Committee on Engine Testing, in arranging forms for the presentation of results, preferred to use such forms for his own tests as should exhibit most clearly the results which he wished to emphasize. With respect to the publication of copies of the original indicator cards, the writer believes he had done as much as is desirable in publishing the combined cards. The publication of copies of the original indicator diagrams for each of the tests would have increased largely the bulk of the paper without at all increasing its value.

The author cannot but express his extreme surprise at the statement of Mr. Barrus, that he had seen some of the original diagrams for the test on engine *A*. These diagrams have not been out of the possession of the writer from the time they were taken, and cannot have been seen by Mr. Barrus. His statement that these diagrams were poor must also be denied. They were seen at the time of the tests, and were found satisfactory both by the Consulting Engineer of the Boston Electric Light Co. and by the engineer representing the engine builders. The engine foundations were very solid, the engine itself very stiff, the indicating gear was a simple photograph device, designed by the engine builders, with but few joints, strong, new, and with no discernable back-lash. The indicators were gone over and oiled every hour. There was no suggestion of shakiness in the cards. The writer considers the indicator cards for this test to be as reliable as can be obtained from engines of this size and speed, and cannot but con-

sider that Mr. Barrus' remarks on these cards, which he has never seen, are either the product of a faulty memory or of a vivid imagination. Copies of the actual cards for the tests on engine *A*, on July 20, 1899, are added to this paper, Figs. 157-160.

The writer finds himself in complete agreement with the re-

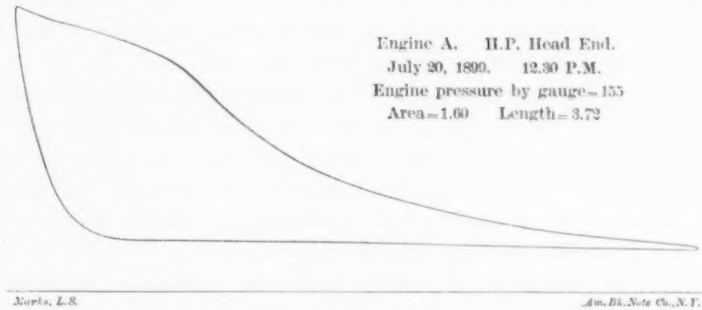


Fig. 157.

marks of Mr. C. V. Kerr. The results of the test of the 5,000 horse-power engine, at the Waterside station in New York, are such as the present tests would show to be probable, unless the reheater were of unusual size and efficiency. The larger the engine the less is the gain to be anticipated by the adoption of the usual devices for securing economy.

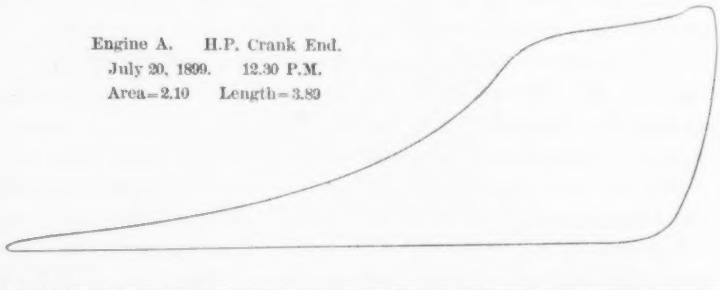


Fig. 158.

Mr. Rockwood's numerous criticisms, although they do not touch the substance of the paper, had perhaps better be answered individually. So complete a series of tests on large size engines of one type has not, the writer believes, been recorded before. They give certain information for which the owners of the engines were willing to undertake the not inconsiderable cost and inconvenience

involved in the making of these tests. The results so obtained have value to all those who have charge of such engines—to all who want to know whether engines of 1,000 to 3,000 indicated horse-power should preferably be run with or without jackets, with or without reheaters, and at what range of loads they may

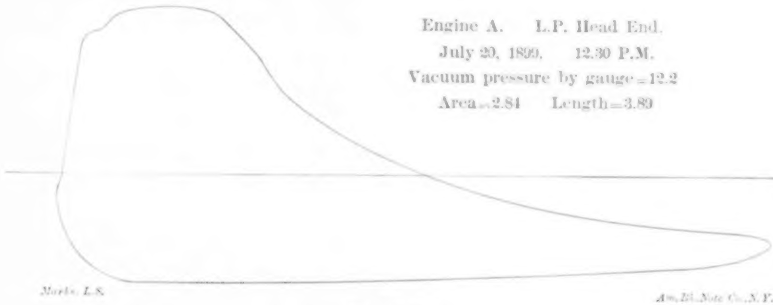


Fig. 159.

be economically run. Such information the writer does not consider to be the threshing of old straw. The information on these points, obtained on small laboratory engines up to 100 indicated horse-power, is of no value for general industrial practice; and it is principally on such small engines that information of the nature indicated has been obtained hitherto.

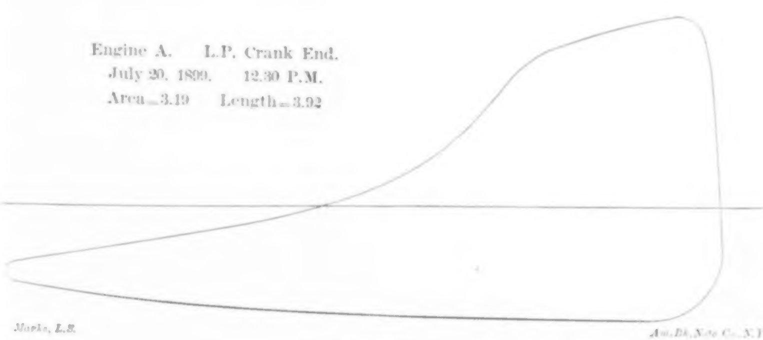


Fig. 160.

Moreover, this paper does not pretend to give "record" results, but only the results of good modern practice. The writer has a high opinion of the possibilities of the high-ratio compound associated with Mr. Rockwood's name, but he would point out that his conclusion with reference to the effect of the size of an engine on

its economy is expressly restricted to engines of ordinary proportions, and intentionally omits a comparison between different types. The writer has no doubt, however, that his general conclusion will be found to apply satisfactorily to a comparison of two of Mr. Rockwood's engines. The comparison instituted by Mr. Rockwood between his own 500 horse-power high-ratio compound and the very different Waterside station engine referred to by Mr. Kerr, can hardly be expected to prove anything.

The writer cannot agree with Mr. Rockwood as to the desirability of having a reheater in the case, where it does not increase the economy of the engine. A separator in the high-pressure exhaust and a readjustment of the cut-off valves will produce a similar result with less trouble and first cost.

With reference to Mr. Rockwood's suggestion that these tests were carried out to gratify the writer's passion for testing steam engines, it may again be noted that these tests were all either acceptance tests, or tests designed by those in charge of the plants to yield certain information as to their performance, without any special reference to the partialities of the writer. The pleasure of working up such tests is perhaps hardly so great as Mr. Rockwood seems to imagine.

No. 1031.*

COMMERCIAL GAS ENGINE TESTING AND PROPOSED STANDARD OF COMPARISON.†

BY WILLIAM P. FLINT, EAST PITTSBURG, PA.
(Member of the Society.)

1. The value of accurate and comprehensive testing of all prime movers, and the determination of a basis of comparison upon which to judge the relative economy of those of any particular class cannot be overestimated.

2. To the manufacturer of heat motors, especially, it is unquestionably necessary, and at the same time most expedient, to incorporate into the scheme of engineering organization an efficient testing department, which shall at once be capable of securing accurate data of the operation of the finished machines without being burdened with any unnecessary refinements. The simplest reliable methods and apparatus are therefore obviously best suited to accomplish the desired results, and it is then left only to determine the most feasible basis upon which to make comparisons of the characteristic data obtained.

3. The friction brake is by long odds the most satisfactory

---

* Presented at the Chicago meeting (May and June, 1904) of the American Society of Mechanical Engineers, and forming part of Volume XXV. of the *Transactions*.

† For further discussion on the same topic consult *Transactions* as follows:

No. 843, vol. xxi., p. 396: "An Efficiency Test of a 125 Horse-Power Gas Engine." C. H. Robertson.

No. 875, vol. xxii., p. 152: "Efficiency of a Gas Engine as Modified by Point of Ignition." C. V. Kerr.

No. 895, vol. xxii., p. 612: "Efficiency Tests of a 125 Horse-Power Gas Engine." C. H. Robertson.

No. 949, vol. xxiii., p. 686: "Temperature of Exhaust Gases." R. H. Fernald.

No. 950, vol. xxiii., p. 705: "Working Details of a Gas Engine Test." R. H. Fernald.

No. 989, vol. xxiv., p. 1048: "Method of Testing Gas Engines." E. C. Oliver.

No. 990, vol. xxiv., p. 1063: "Performance of an Internal Combustion Engine Using Kerosene and Gasolene as Fuel." Halladay and Hodge.

No. 991, vol. xxiv., p. 1095: "Test of a 12 Horse-Power Gas Engine." C. H. Robertson.

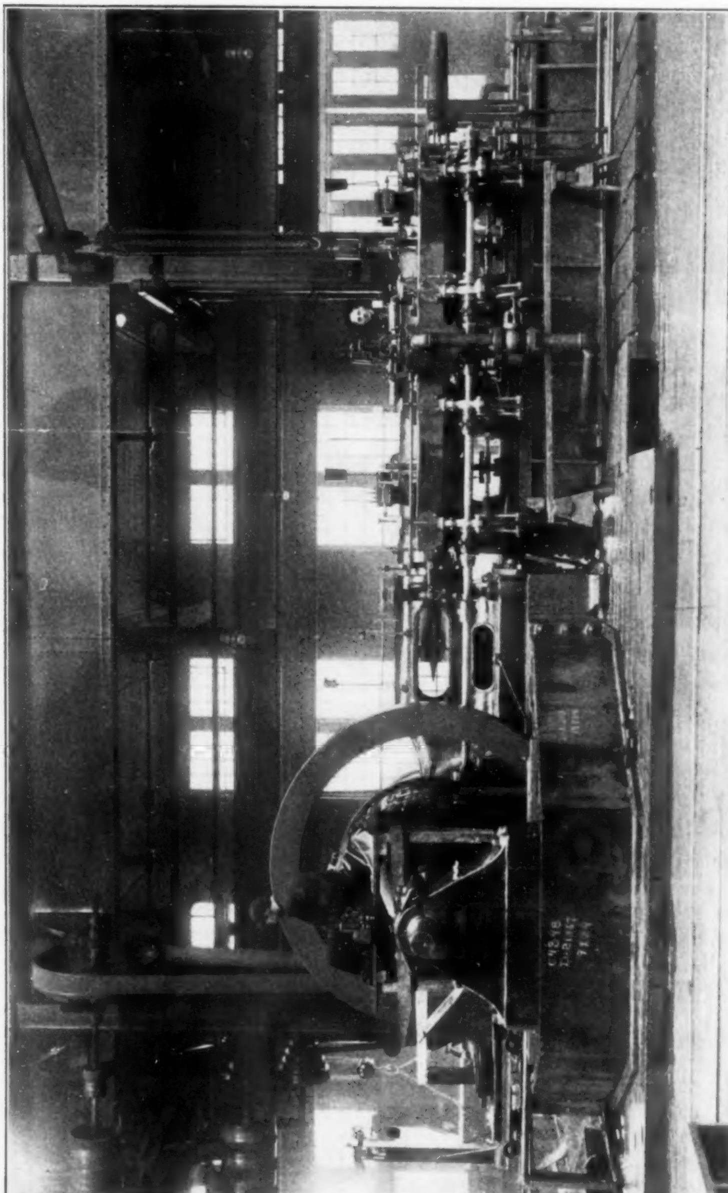


Fig. 161.

means of measuring the power developed. Its accuracy depends on the following factors:

First. The accuracy of standard platform scales. This is easily verified by standard weights.

Second. Allowance for the weight of the blocking on the scales and the unbalanced weight of the brake band and arm. These are both easily determined by actual weighing.

Third. The brake radius, or distance from center of the shaft to the knife edge supporting the brake arm.

Fourth. The revolutions per minute.

Fifth. The steadiness with which the scales are kept balanced.

4. If we compare the above simply verified facts with those required by an indicator test, it will be at once apparent how much superior the brake is to the indicator as a means of quickly and reliably measuring power. Then also the available or brake horse-power, and not the indicated horse-power, is the thing for which engines are run.

The Friction Brake:

5. The brake is applicable to large as well as small engines, as may be seen by reference to Figs. 161 and 162, showing respectively a 300 brake horse-power double-acting tandem and a 25 brake horse-power single-acting vertical gas engine undergoing test. Fig. 163 shows several engines set up in readiness for shop test.

6. In practice it is necessary to lubricate the brake wheel for the purpose of making the friction uniform, and to prevent the wooden cleats becoming locally heated and taking fire. It is found that strips of fat salt pork make the best lubricant, for they slowly fry and thus keep the wheel uniformly and continuously greased. The wheel must, of course, be cooled, and this is usually provided for by casting re-entrant lips on it, which will retain a layer of water by virtue of centrifugal force when the brake wheel is in motion.

7. A stream of water is fed into the brake wheel at one point and scooped out by a stationary pipe at another point. By this means the temperature of the brake wheel is controlled.

8. When the temperature and lubrication of the brake wheel are kept constant, it is not difficult for a man with a wrench to keep the brake-band tension screw so adjusted that the scale beam of the platform scales is always substantially balanced. If the load becomes too great, he slackens the tension, and if too small, he increases it.

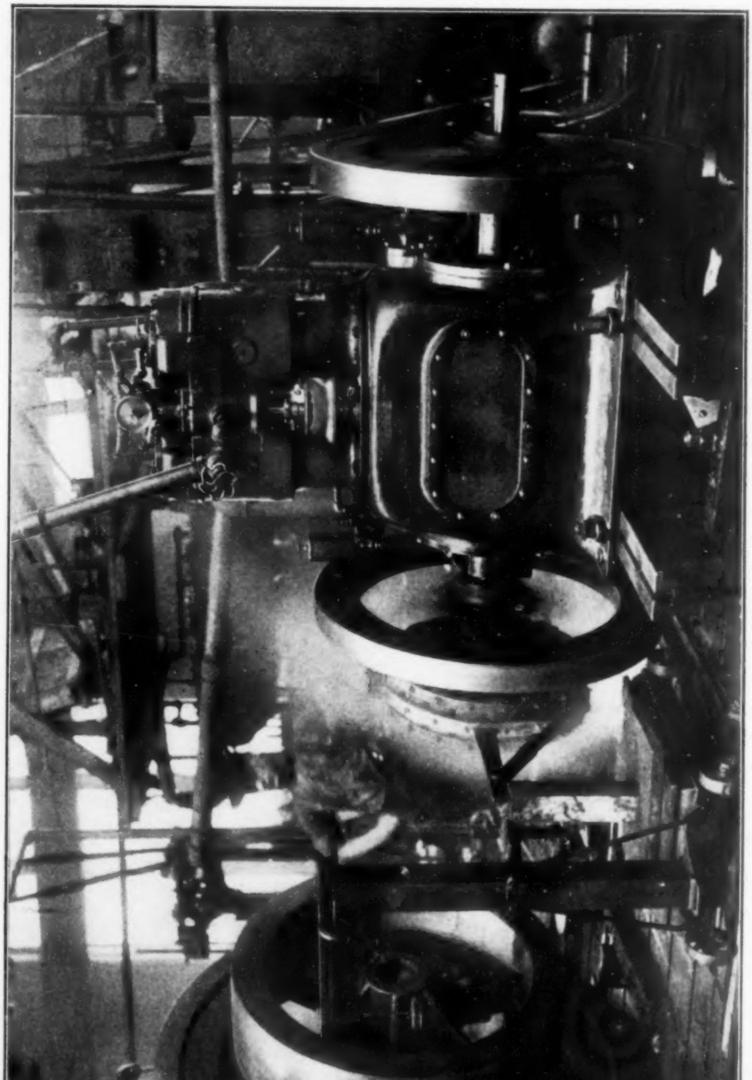


FIG. 162

9. Occasionally considerable annoyance may be experienced by sudden changes of friction occurring which are due to a little water getting on to the outside of the brake wheel. This seems to chill the lubricant and have a disturbing effect out of all proportion to the cause. It is a small practical point well worth attention, for with irregular variations in the friction, the man at the brake cannot keep the load constant. The remedy is very simple—keep water off the outside of the brake wheel.

The Indicator Card:

10. The indicator, while not as useful for power determinations, is of great value for showing what is going on in the cylinder, and cards should always be taken in connection with the brake test. It will often be superfluous to calculate the indicated horse-power from them. If engineers in charge of gas engines made more frequent use of the indicator, they would frequently be able to obtain increased satisfaction by detecting faulty adjustments, particularly of the point of ignition.

Engine Speed:

11. Where an engine is being tested on a constant load and quality of gas, the governor should hold the speed constant enough to warrant the use of a good make of hand-speed counter.

Where the above condition is not met a continuous counter must be employed.

Gas Measurements:

12. A good meter, whose accuracy is checked occasionally, by means of prover tests, over the range for which it is used, is the best instrument for this purpose. The temperature and pressure of the gas at the meter and the barometer reading (uncorrected for sea level) are needed for correcting the meter readings to standard conditions.

Calorimeter determinations or chemical analyses will give the heat value of the gas.

Comparison of Tests:

13. After tests are made we still desire to know what they mean. We must have a standard with which to compare the results obtained on a given engine and as a basis for predicting what that engine ought to do.

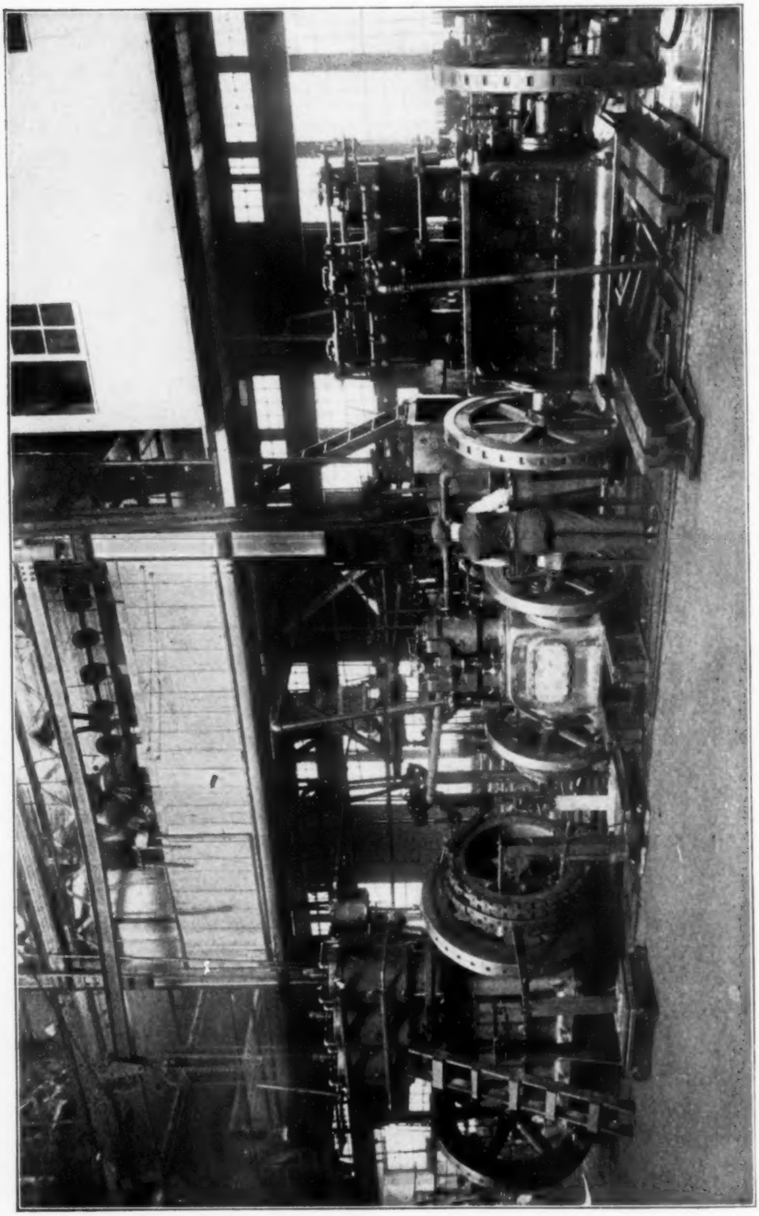


Fig. 163

TABLE I.

DATA AND CALCULATIONS FOR FIGURE (5).

16½ x 24 Gas Engine—Test No. 904.

Type—Horizontal, double-acting, tandem, single crank, four cycle throttling engine.

Diameter of cylinders	16½ inches.
Stroke..	24 "
Diameter of piston rod (in three explosion chambers).......	5⅝ "
Diameter of tail rod (in one explosion chamber)............	2⅞ "
Full load test speed.....................................	181 R.P.M.
Number of charges per two revolutions.....................	4
Suction displacement per minute..........................	981 cubic feet.
Reduction factor $= \frac{345 \text{ cubic feet}}{981 \text{ cubic feet}} =$	0.352

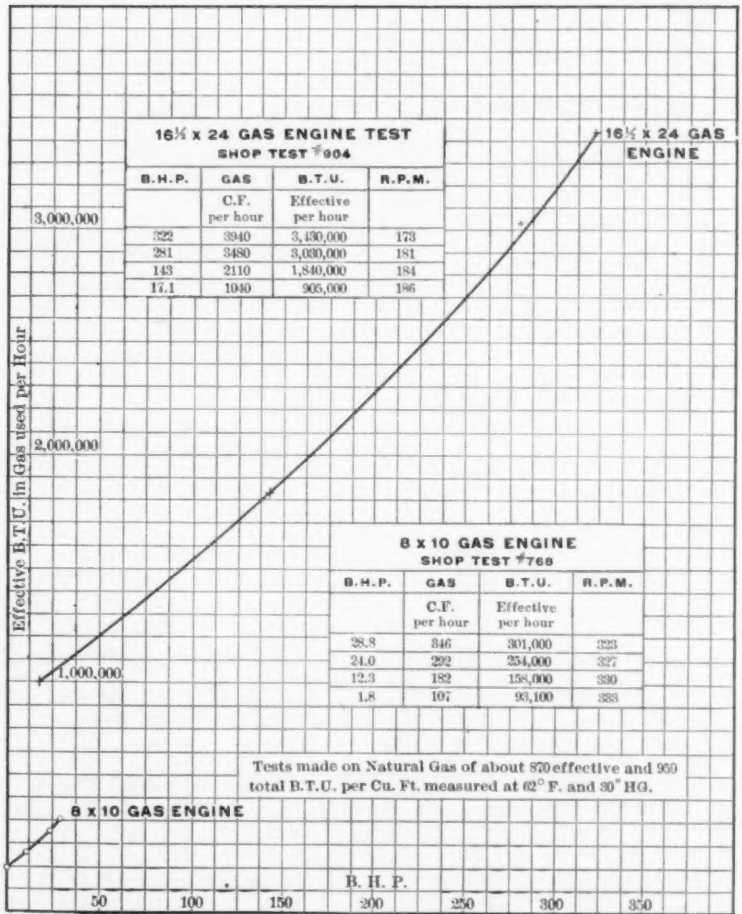
Actual Readings.	Readings Calculated to 100 Horse-Power Basis.
322 B. H. P. 3,430,000 B. T. U.	113.3 B. H. P. 1,207,000 B. T. U.
281 B. H. P. 3,030,000 B. T. U.	99.0 B. H. P. 1,066,000 B. T. U.
143 B. H. P. 1,840,000 B. T. U.	50.3 B. H. P. 646,000 B. T. U.
17.1 B. H. P. 905,000 B. T. U.	6.0 B. H. P. 318,000 B. T. U.

NOTE.—The second column of figures are obtained by multiplying those in the first column by the reduction factor 0.352.

14. The most natural standard is the cubic feet of gas used per brake horse-power hour. When, however, we attempt to use this, we find that for each engine it is a function of the load carried. Its value varies considerably at full load, and very rapidly at light loads, until at no load it reaches infinity. We must make several tests at different loads in order to get the law of this variation. When we calculate the gas consumption per brake horse-power hour at light loads, we find that a very small variation in the total amount of the gas used, or of the load at which the test is made, makes a very large and unmeaning variation in the cubic feet per brake horse-power hour.

15. This unmeaning characteristic of the gas per brake horse-power curve at light loads leads one to prefer the use of a curve plotted with the brake horse-power as abscissæ, and the cubic feet of gas, or better yet the British thermal units of gas, per hour as ordinates. Fig. 164 shows two such curves, one for an 8 by 10,

and the other for a $16\frac{1}{2}$ by 24 gas engine. These two curves, however, are so different in size that they are incomparable with each other, even though the engines require about the same number of British thermal units per brake horse-power hour at corresponding



16½ x 24 GAS ENGINE TEST
SHOP TEST #904

B.H.P.	GAS	B.T.U.	R.P.M.
	C.F. per hour	Effective per hour	
322	3940	3,430,000	173
281	3480	3,030,000	181
143	2110	1,840,000	184
17.1	1040	905,000	186

8 x 10 GAS ENGINE
SHOP TEST #768

B.H.P.	GAS	B.T.U.	R.P.M.
	C.F. per hour	Effective per hour	
28.8	846	301,000	323
24.0	292	254,000	327
12.3	182	158,000	330
1.8	107	93,100	333

Fig. 164.

loads. They need to be plotted to such scales that they may be readily compared. Without replotting, however, they give at a glance the British thermal units required on each engine at any load, and from them the British thermal units per brake horse-power hour curves may be readily derived.

TABLE II.

DATA AND CALCULATIONS FOR FIGURE (5).

8 x 10 Gas Engine—Test No. 768.

Type—Single acting, two cylinder, four cycle, vertical throttling engine.

Diameter of cylinders	8 inches.
Stroke	10 inches.
Full load test speed	327 R.P.M.
Number of charges per two revolutions	2
Suction displacement per minute	95.2 cubic feet.
Reduction factor $= \frac{345 \text{ cubic feet}}{95.2 \text{ cubic feet}} =$	3.62

Actual Readings.	Readings Calculated to 100 Horse-Power Basis.
28.8 B. H. P. 301,000 B. T. U.	104.0 B. H. P. 1,090,000 B. T. U.
24.0 B. H. P. 254,000 B. T. U.	87 B. H. P. 920,000 B. T. U.
12.3 B. H. P. 158,000 B. T. U.	44.5 B. H. P. 573,000 B. T. U.
1.8 B. H. P. 93,100 B. T. U.	6.5 B. H. P. 337,000 B. T. U.

NOTE.—The second column of figures are obtained by multiplying those in the first column by the reduction factor 3.62.

Proposed Basis of Comparison:

16. The maximum power of a given gas engine depends on the number of British thermal units it can take in per minute, and on the percentage of this heat which it can turn into brake horse-power.

17. With engines of about 10 brake horse-power per cylinder and larger, there is but little variation in the efficiency of similar engines, which may be attributed to the size. The consequence of this is that the power of gas engines of the four-stroke cycle type varies almost in proportion to their suction displacement, and some one size of engine may be taken as a standard to which to reduce the figures obtained on other engines. The curves thus obtained from the results of tests on many sizes of engines are mutually comparable. They furnish a basis for predicting what still other sizes of engines, not yet tested, may be expected to do.

18. The results of tests on a large number of similar engines using natural gas for fuel have shown that every 345 cubic feet mixture displacement per minute will give in the neighborhood of 115 maximum brake horse-power, or 100 rated brake horse-power.

19. The above displacement is figured from the area of the piston, length of stroke, and the greatest number of charges per minute at full load speed, and furnishes the data upon which it is proposed herein to establish a basis of comparison.

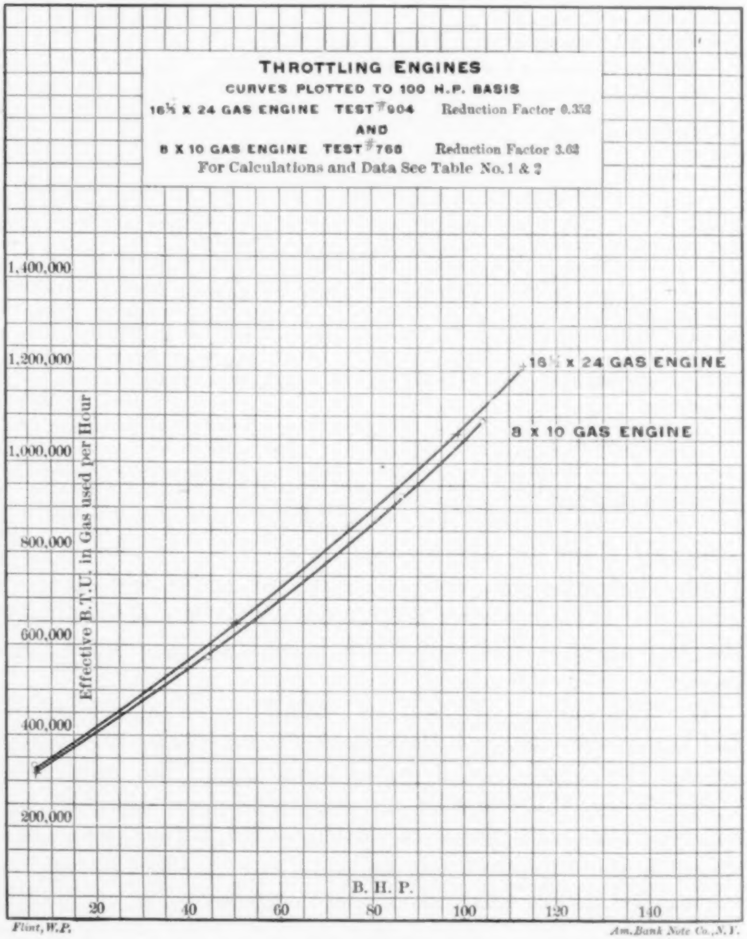


Fig. 165.

20. Fig. 165 shows the two curves given in Fig. 164 replotted to this new basis. The necessary calculations are very simple, and are given in tables No. 1 and No. 2.

21. The British thermal units per brake horse-power hour for

TABLE III.

DATA AND CALCULATIONS FOR FIGURE (6).

7 x 10 Gas Engine—No. 1307.

Type—Single acting, single cylinder, four cycle, horizontal, hit and miss engine.

Diameter of cylinders	7 inches.
Stroke	10 inches.
Full load test speed	253 R.P.M.
Number of charges possible per two revolutions	1
Suction displacement per minute	28.2 cubic feet.
Reduction factor $= \frac{345 \text{ cubic feet}}{28.2 \text{ cubic feet}} =$	12.23

Actual Readings.	Readings Calculated to 100 Horse-Power Basis.
5.22 B. H. P. 90,500 B. T. U. 250 R. P. M.	63.8 B. H. P. 1,108,000 B. T. U.
3.35 B. H. P. 63,600 B. T. U. 253 R. P. M.	41 B. H. P. 778,000 B. T. U.
2.60 B. H. P. 55,200 B. T. U. 255 R. P. M.	31.8 B. H. P. 675,000 B. T. U.
0 B. H. P. 32,400 B. T. U. 256 R. P. M.	0 B. H. P. 396,000 B. T. U.

NOTE.—The second column of figures are obtained by multiplying those in the first column by the reduction factor 12.23.

any load may be obtained by dividing the British thermal units reading by the corresponding brake horse-power reading. This is particularly simple at the 100 horse-power load when it simply means pointing off two places of the British thermal units reading.

22. Tests plotted in this way show up graphically the relation between the power developed per unit of mixture displacement in different engines, as well as the relative economy in gas consumption.

23. By plotting in this manner, the results of a large number of tests of gas engines, a manufacturer can ascertain the characteristic British thermal units brake-horse-power hour curve for each size and type, and from these can predict what new sizes should do. Experience teaches that when reduced to a common basis, there is but little more variation between the curves for similar engines of different sizes than between individual engines of the same size. There are so many factors influencing the exact loca-

tion of the curves in question, that under commercial conditions, a good many tests have to be run before safe average curves can be drawn. For this reason it is most helpful to be able to readily and intelligently compare the results of different sizes, to the end that the experience gained on a size of which many engines have

TABLE IV.

DATA AND CALCULATIONS FOR FIGURE (6).

6 x 7 Gas Engine—No. 1141.

Type—Single acting, single cylinder, four cycle, horizontal, hit and miss engine.

Diameter of cylinder	6 inches.
Stroke	7 inches.
Full load test speed	306 R.P.M.
Number of charges possible per two revolutions	1
Suction displacement per minute	17.52 cubic feet
Reduction factor $= \dfrac{345 \text{ cubic feet}}{17.52 \text{ cubic feet}} =$	19.7

Actual Readings.	Readings Ca'culated to 100 Horse-Hower Basis.
3.76 B. H. P. 770,000 B. T. U. 305 R. P. M.	74 0 B. H. P. 1,515,000 B. T. U.
2.90 B. H. P. 698,000 B. T. U. 306 R. P. M.	57.3 B. H. P. 1,375,000 B. T. U.
1.36 B. H. P. 513,000 B. T. U. 316 R.P.M.	26.8 B. H. P. 1,010,000 B. T. U.
0 B. H. P. 357,000 B. T. U. 327 R. P. M.	0 B. H. P. 703,000 B. H. P.

NOTE.—The second column of figures are obtained by multiplying those in the first column by the reduction factor 19.7.

been built may be utilized in criticising the performance of a new size.

24. The average curves thus determined may, with good advantage, be replotted in a manner similar to Fig. 164 for the use of agents who need to be able to quickly answer questions as to the total amount of gas which a given engine will use at each of several different loads.

25. Fig. 166 shows the results obtained on two small, "hit and miss" gas engines plotted to the 100 horse-power basis, and is given to illustrate the effectiveness of the graphical com-

parison of tests on engines showing very different efficiency and capacity per unit of mixture displacement. These particular engines have low compression, which has a good deal to do with the poor performance shown.

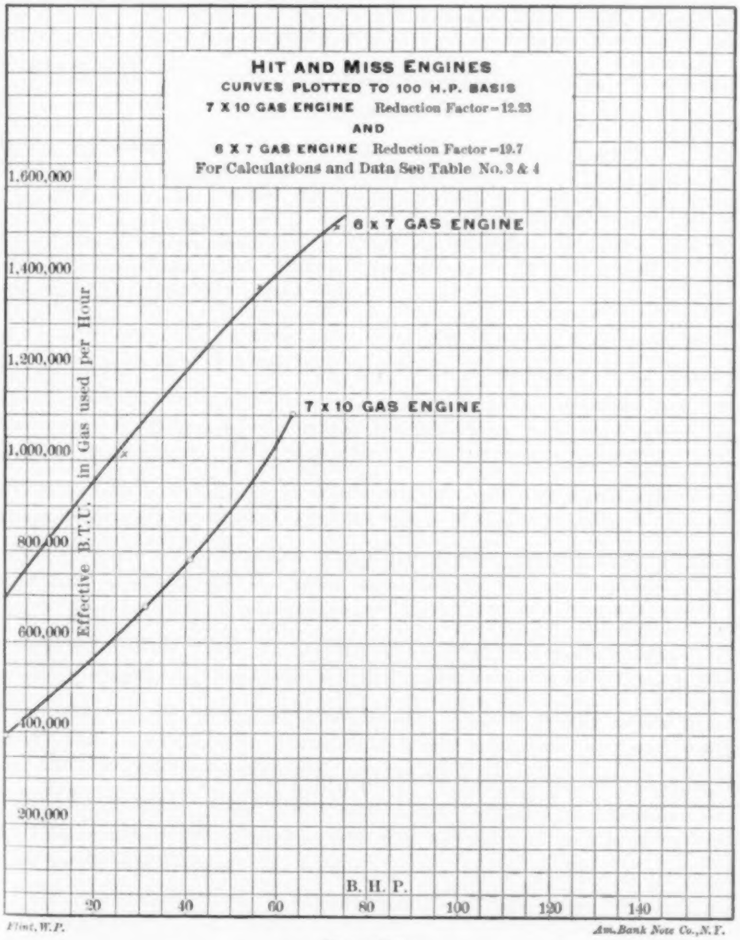


FIG. 166.

26. Besides enabling an average performance curve to be located with considerable certainty, this 100 horse-power basis method lends itself well to the determination of limiting curves and figures between which the performance of any engine may be expected to lie. This is often very useful in drawing atten-

tion to defects in an individual engine, and in detecting errors which occasionally may be made in shop tests. It should prevent defective engines from being sent out, and also prevent unusually good or poor records being entered without the careful verification which they should receive.

Relation of Mixture Displacement to Power and Efficiency of the Engine.

27. It is interesting in this connection to do a little guessing as to what factors are involved in the above relation for the standard 100 horse-power engine here proposed.

28. It will, by assumption, have a nominal mixture displacement of 345 cubic feet per minute at full load speed and a maximum power of 115 brake horse-power. The overload speed will be less than that at full load, and if we assume it 2 per cent. less, we shall have 98 per cent. of 345 or 338 cubic feet actual displacement per minute at overload. The exhaust gases in the clearance spaces are often at a slight pressure at the end of the exhaust stroke, and have to expand to atmospheric pressure before any new charge is sucked in. Again, there is usually a slight vacuum at the end of the suction stroke. Light spring stop motion cards taken to show plainly the suction and exhaust lines indicate that there may be a loss of about 5 per cent. of the displacement volume. Then 95 per cent of 338 cubic feet equals 321 cubic feet, and gives the actual suction displacement filled with new gases per minute. This 321 cubic feet of mixture is measured at a temperature above that of the atmosphere, due to the heat absorbed from the cylinder walls and to the admixture of hot gases left in the clearance space from the previous explosion.

29. It must be admitted that the author has no data at hand for arriving at a fair figure for this temperature, other than that of assigning it such a value as will make the final result seem reasonable. By assuming this temperature to be 210 degrees Fahr., or 671 degrees Fahr. absolute, the correction becomes $\frac{523}{671} = 78$ per cent., which means that 250 cubic feet of mixture per minute at standard pressure and temperature are drawn in.

30. The next question is the number of heat units which this 250 cubic feet of mixture contains. To simplify the calculation consider the gas to be all CH_4. This is so nearly true of natural gas that the results are applicable also to it.

$$CH_4 + 2\ O_2 + 7.50\ N_2 = CO_2 + 2\ H_2O + 7.50\ N_2$$

is the equation for the theoretical explosive mixture, but we must have an excess of air present or some gas will not be burned. Here again it is necessary, in the absence of measurements of the volume of air used, to resort to an arbitrary assumption and add 10 per cent. excess of air. The equation then becomes

$$CH_4 + 2.2\ O_2 + 8.25\ N_2 = CO_2 + 2\ H_2O + 8.25\ N_2 + 0.2\ O^2 \text{ or}$$
$$CH_4 + 10.45 \text{ air} = CO_2 + 2\ H_2O + 8.25\ N_2 + 0.3\ O_2$$

1 cubic foot CH_4 gives 1,015 total British thermal units, or (allowing for the heat of vaporization of water formed which is not available for power purposes in any gas engine) 915 effective British thermal units per cubic foot of gas. 1 cubic foot gas + 10.45 cubic feet air contains 915 effective British thermal units, or we have $\frac{915}{11.45} = 80$ effective British thermal units per cubic foot of mixture. Thus the 250 cubic feet of mixture per minute will contain $80 \times 250 = 20,000$ effective British thermal units.

31. The assumption at the start of this calculation was that the engine would give 115 maximum brake horse-power, and as 42.4 British thermal units per minute is the theoretical equivalent of 1 brake horse-power minute, this means that $115 \times 42.4 = 4,870$ British thermal units out of 20,000 British thermal units per minute are transformed into brake horse-power; that is, the British thermal unit equivalent of brake horse-power divided by effective British thermal units in the mixture equals $\frac{4,870}{20,000} = 24.35$ per cent. efficiency.

This efficiency means 10,430 effective British thermal units per brake horse-power hour, and is what is obtained in practice under good conditions.

32. It is not claimed that this calculation represents the results of particular tests. It is given only to draw attention to the factors which are involved in the relationship existing between the displacement, power and efficiency of the engine here assumed as a basis for comparison.

33. Incidentally this reasoning indicates a line of laboratory experiments which may throw light on the temperature of the gases at the beginning of compression, since that is the only factor in the series which cannot be quite readily measured.

34. This system of plotting the results of gas engine tests to a common basis has proved so helpful in the study of one well-known make of gas engine, and is so simple of application, that it is commended to the attention of all engineers interested in this subject, and it is hoped that those who publish tests on important engines will take readings on several different brake loads, including the maximum, and will give the few easily determined facts about the engine which are needed by those who wish to plot the data as herein proposed.

DISCUSSION.

Prof. Storm Bull.—The paper of Mr. Flint has interested me very much, especially as it shows the methods of testing gas engines used by the Westinghouse Machine Co. in their shops. That an engine should be tested for various loads and not alone for full load is very true, and it seems very instructive to put the results in the manner indicated by the author: the brake horse-power as abscissæ and the British thermal units consumed as ordinates.

However, when the author attempts to establish a standard with which he proposes to compare all other engines, it would seem that he has been very unfortunate. It may be true enough that the efficiency of similar engines varies but very little for sizes above 10 horse-power, provided, however, that the quality of fuel is the same in all cases. But in making such an assumption the implication is that the efficiency has already now reached a maximum beyond which one could not hope to come. This certainly is not true, and consequently the standard selected is at the most a temporary one, and one which must be changed with the advance in the efficiency of the gas engine.

But the most serious objection to the selection of the standard for comparison is that it is based upon the consumption of a certain number of cubic feet of natural gas per horse-power and per minute. This number will be a different one for all other kinds of gas, and even for the various grades of natural gas, and for two reasons: first, because of the different number of British thermal units contained in one cubic foot of the various gases, and, secondly, because the compression must of necessity be a different one for each one of the gases, upon which again the efficiency depends. From this it will be seen that only engines using the same kind of gas can be compared in the manner

proposed by Mr. Flint; there would have to be as many standards as there are kinds of gas for which engines are being made in the factory.

*Mr. Edward J. Chambers.**—I almost think I ought to apologize for inflicting a few words upon you this morning, but you are such vampires for work in America that I don't know how to keep pace with you. I think the next thing I shall propose will be that the heads of these two associations shall not exceed the age of twenty-five, because it is hard for an old man to keep up with the rush.

The paper we have listened to is extremely interesting to me. It is the commercial gas engine. I think our thanks are really due to the author. At the same time I do not think he has gone far enough. I am a thorough convert to the use of gas engines myself. When I hear you gentlemen wasting your time discussing the relative merits of steam engines and electric motors, I really pity you. Some ten years ago (I have had the good fortune to live a long time, with its attendant misfortune also) I had the good fortune to have to lay out my third large works in my engineering career. It has not been only to lay them out and pocket my fees and go away, but lay them out and work them afterwards and make money out of them. My first experience was putting down big boilers in the center of the work and running my shafting all over the place. My second experience came some years afterward, when I thought I wouldn't carry the shafting all over the place, but I would carry the steam. So I had my big boilers and I carried the steam all over the works. If you know anything about steam (and probably you do), and of the small steam engine, you will quite understand that I very soon had enough of that. Then I had one of the greatest good fortunes that ever fell to the lot of an engineer, and that is to be able to design and build a third works and fully equip it, and if that isn't right then I think the designer ought to be voted absolutely no good. I very carefully went into steam then and into electricity. I was delighted with electricity, but then I had a cheque book and only a certain amount of balance in the bank, and consequently I didn't go in for electricity. For some of you gentlemen who live near Niagara I can quite understand your adopting electricity, and I can quite understand your adopting it under many other conditions. I can quite understand, also, why you gentlemen who

* Member of the Institution of Mechanical Engineers of England.

live next door to, and can tap on the ground and get, natural gas adopt it.

But let me come to this paper, where, as I said before, I do not think the author has gone far enough. He first of all assumes that the gas is going to be all regular town gas. Had I been compelled to use that let me tell you I shouldn't have adopted gas engines, but I have had to make the gas, and there are various ways of making it. I adopted the Dowson gas plant some ten years ago; I started with three small engines. Don't ever make that mistake. Have your engines big enough. I determined I would go in for gas engines. I started and tried in a little way at first, but I had to double my power and take the old ones out. Indeed, I had to do that twice, but that wasn't entirely my fault. You see, I had some partners, and my partners wouldn't go the whole hog. Those cautious partners are a great nuisance sometimes. But I got my big engines after all. The three engines run up to about 230 horse-power. I felt then that I must do exactly what Mr. Flint did, but he talks of it more especially as the *maker* of gas engines, and from the calculations that our friend, Mr. Mathot, has brought here, I have no doubt that he too looks at it as the manufacturer of gas engines. But when I caught sight of that word "commercial," I said, "This is business," and I shall find some way where I as a user can learn something. It is suggested to us that brake tests should be made. Now, you know perfectly well that we have something else to do besides putting brake tests on. The only brake tests which I ever put on is work in the shop. Therefore I have got to fall back on the indicator diagrams mainly for my own test of the engines. I have an indicator outfit attached to every engine, and my rule is to take a diagram during the run of four hours, when the full load in on.

What I want is that you gentlemen should devote yourselves to gas engines, because I tell you you may do wonders with electricity, but at the present time gas engines hold the field. After the first twelve months I had made up my mind to test in every possible way what the result was. At the end of the first twelve months I had my accountant make up for me a paper showing how they compared with steam, and I received a most extraordinary result; I thought it couldn't be true, and I went over the matter myself and checked every item to make sure they were not humbugging me, and I found out that my costs had gone

down about 50 per cent., and felt prosperous, and you know prosperity often does us harm. One of the worst things that a nation can have or an individual is a long run of prosperity, and this run of prosperity stopped me investigating, on the lines Mr. Flint has followed. Now when I go home, in spite of what little prosperity I have had, I will go in for gas engine tests, and I should like to get into correspondence with Mr. Flint, and if I can do anything to add to the information of the members of our brother engineering Society here, I shall be only too delighted to give you the benefit. I shall go in for having some way of testing the gas so that I can turn on the tap, get a vessel full and know exactly what the composition of the gas is, because that is the first point, getting your gas right. I will do the best I can, and I will let the members of this Association have the full benefit of anything I may be able to accomplish.

Professor Jacobus.—The standard of comparison proposed by the author of the paper, where the heats of combustion of the gas used per hour are plotted as ordinates and the brake horse-powers as abscissae, is equivalent to what is often done in steam engine tests where the water consumptions per hour are made the ordinates and the indicated or brake horse-powers the abscissae. In either case a curve is obtained which approaches much more nearly to a straight line than the corresponding curve of heat or water consumption per hour per horse-power, and for this reason it can be plotted with greater accuracy when there are but few experiments than the corresponding curve of heat or water consumption per hour per horse-power. There is as much to be gained by plotting this curve for a gas engine as in plotting the corresponding curve for a steam engine, and the author does well in bringing the method forward and pointing out its advantages.

In computing the heat consumption the latent heat of the water vapor contained in the products of combustion has been deducted. The efficiency based on this figure is greater than that based on the total heat of combustion which includes the latent heat of the vapor. In the report of the Committee of this Society to codify and Standardize the Methods of Making Engine Tests it was recommended that the total heat of combustion be employed in computing the efficiency. The committee considered this matter very carefully, as they were aware that by recommending the use of the total heat of combustion they would be

at variance with what has been done by many prominent engineers. They finally decided that it was no more logical to subtract the latent heat in the water vapor in the products of combustion of an internal combustion engine than it would be to subtract the latent heat of the exhaust steam of a steam engine. Suppose we have a gas engine in which the fuel is hydrogen gas, and instead of supplying air to burn this gas we supply oxygen. This would give an internal combustion engine in which the working fluid would be highly superheated steam, and it would be no more logical to deduct the latent heat of the vapor in the exhaust of such an engine than it would in any other engine using steam. Again there are some internal combustion engines where water is injected and the working fluid is thereby made to contain considerable vapor. If the exhaust of such an engine is say one-half water vapor, the fact that it is illogical to deduct the latent heat of the vapor in the exhaust is made more apparent than where the amount of the vapor is small. Any vapor produced in combustion is part of the working fluid and does its proportion of useful work in the cylinder; the latent heat contained in the exhaust should therefore not be deducted from the heat supplied to the engine in computations of efficiency. In the report of the Committee of this Society on Boiler Tests it is recommended that the total heat of combustion be used in computing the boiler efficiency, and the recommendations of the two Committees of this Society bearing on this point are therefore the same.

It is stated in the paper that a stream of cooling water is fed into the brake wheel at one point and is scooped out by a stationary pipe at another point. This method is objectionable if there is a great quantity of cooling water circulated, as there may be a considerable reaction produced against the pipe used for scooping out the water, which, together with the force required to accelerate the water fed to the wheel, will not be recorded on the scales of the brake. The readings for the load on the brake, will, therefore, be lower than they should be. The best plan to adopt in most cases, and that recommended in the report of the engine test committee already referred to, is to feed just enough water to the brake wheel to make up for that lost by evaporation. This will ordinarily keep the temperature low enough, and it is a more simple way of cooling the wheel than the one described in the paper.

It is sometimes desirable to place a brake on a wheel already on an engine, and which is not provided with internal flanges for holding the cooling water. When this is done the wheel can often be cooled by allowing small streams of water to strike the inside and the outside of the rim. In such cases it is best to use a rope brake as this is the least affected by the action of the water. The brake must be arranged to lift a dead weight so that when the rope tends to bind fast to the wheel the weight will be raised, and the tension on that part of the rope which passes around the wheel thereby diminished. The method of using such brakes is illustrated in the report of the Engine Test Committee already referred to. We have frequently used the cross lever device therein described with good success. In one case where the power was taken from an ordinary cast iron fly-wheel of over 12 feet in diameter, and the net load on the brake was over one ton, the brake gave steady readings in tests of more than ten hours duration.

Mr. Rudolphe Mathot.—I must first of all apologize for my bad English, but as I am a stranger I hope you will excuse me, and if I do not use the right word allow me to invent one for the purpose.

You have heard the valuable communications and discussions concerning the steam turbine and steam engine. In fact, they are two serious competitors. But both steam engines and turbines have something more than a competitor; they have a common opponent, the gas engine, of which all the partisans of steam engines and turbines seem to be afraid, quite as if they had to deal with a terrible enemy which it is better not to talk about.

In Europe the importance of gas engines has become so great during these last four or five years, since they have been supplied from blast furnaces and special producers working in the factory, that there are about 200,000 brake horse-power on work from 5 up to 2,000 brake horse-power.

And this leads me, as a contribution to Mr. Flint's paper to ask to be allowed to read to you a report which shows remarkable figures of a test upon a modern gas engine. I may say that you may be quite confident with respect to the truth of my figures. Of course, I am not a business man, but an independent specialist, having made a specialty for ten years of testing all kinds of gas engine.

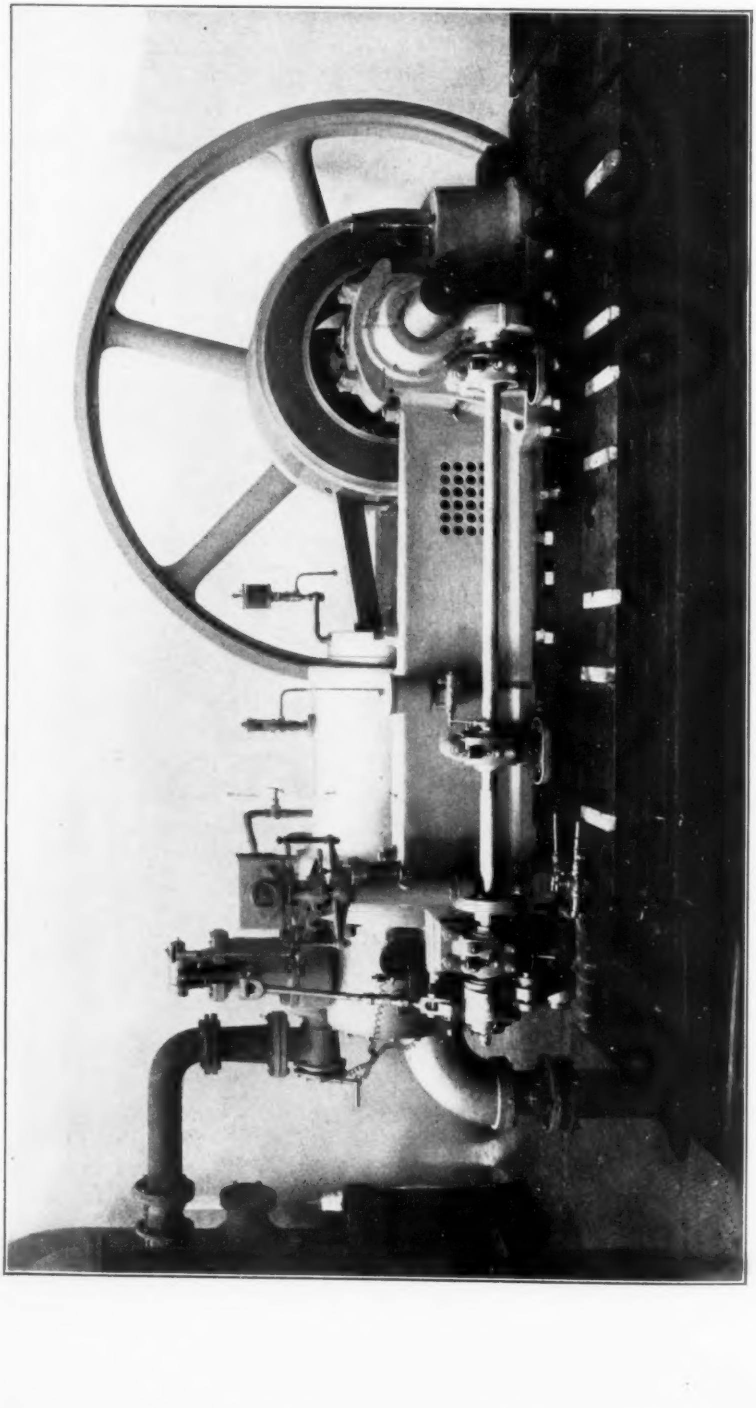


FIG. 167.—OTTO SUCTION PRODUCER GAS ENGINE.

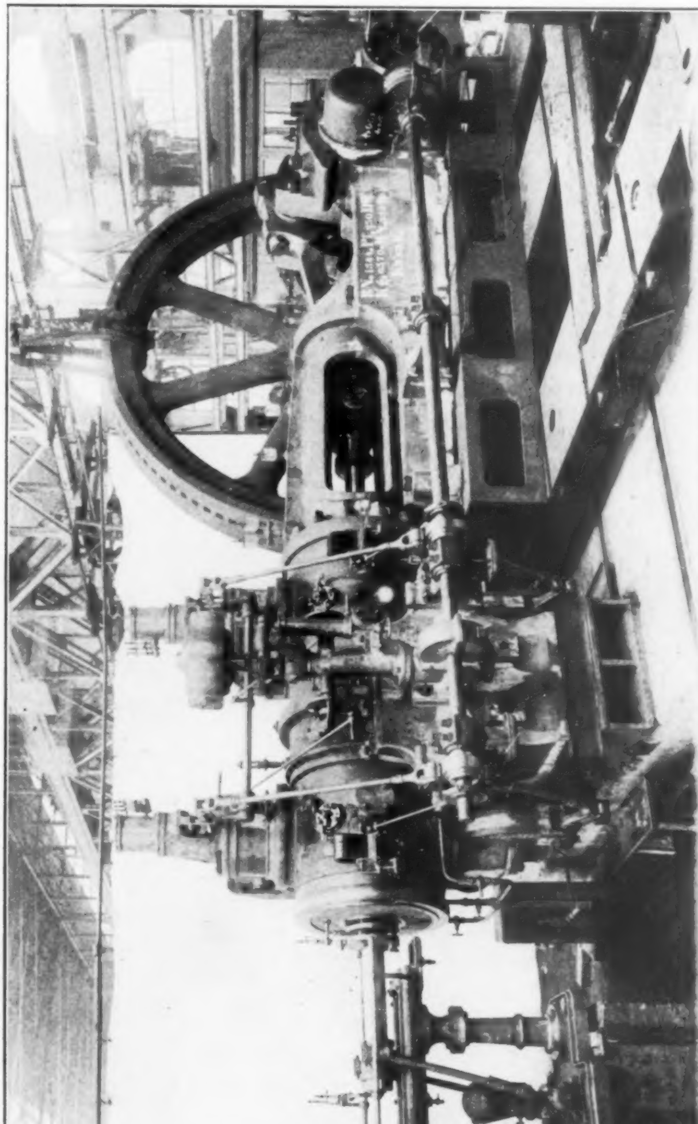


Fig. 168.—Otto Suction Producer Gas Engine.

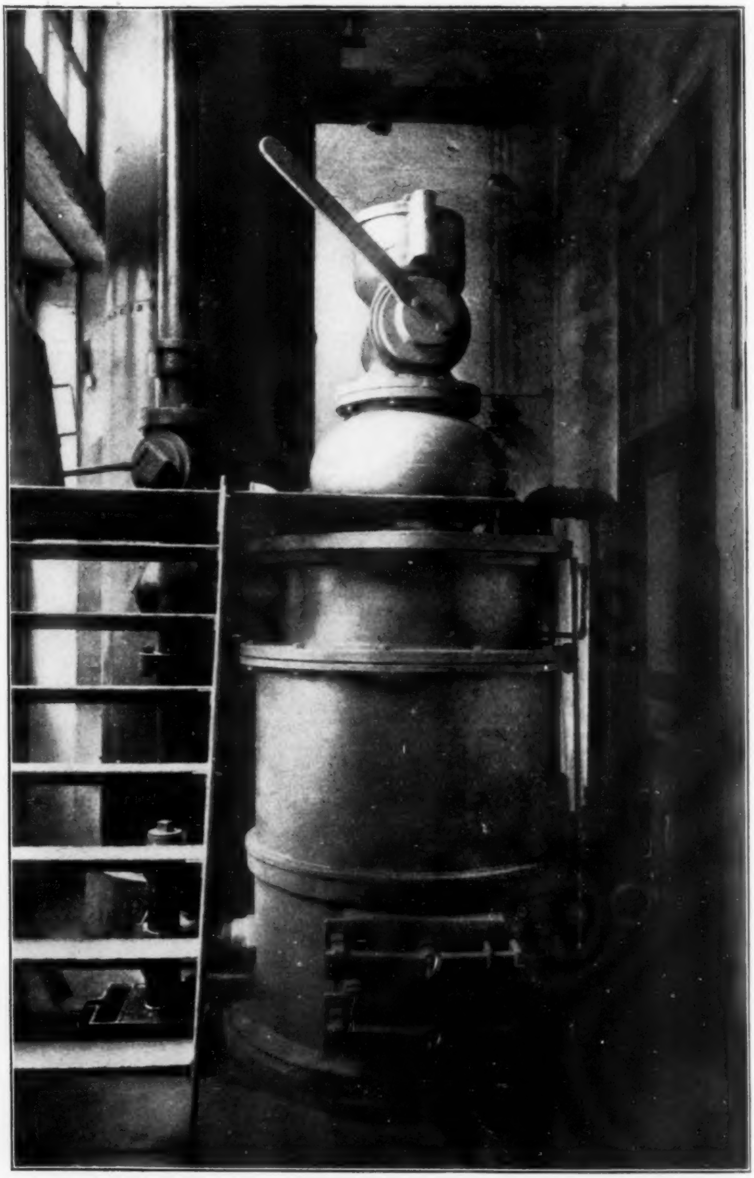


FIG. 169.—60 H. P. OTTO SUCTION PRODUCER.

REPORT.

At the request of Mr. G. Stein, Chief Engineer of the GASMOTOREN FABRIK DEUTZ, at Cologne, Germany, we have tested a gas engine of 60 brake horse-power supplied by a gas producer working by suction.

The apparatus was installed in the testing shop at the above firm's works.

The object of the test was to ascertain the working conditions and the fuel consumption of the producer.

The engine was one of their model G 9, horizontal, single cylinder with valves, working after the Otto cycle. The ignition was magneto electrical, and a heavy fly wheel was fitted with outer bearing.

The governor acted on the stroke of the lever of the inlet valve by the displacement of its support. This realized the regulation by the admission of a "variable quantity of mixture at constant rate."

The generator was of the "suction" type with internal vaporiser and overflow discharged in the ashpit.

The cooling and washing of the gas was done in a simple coke scrubber.

The gas pipe ended in an expansion tank before the inlet to the engine.

A balanced Prony brake with water circulation was fitted on the crank shaft and a speed register recorded the total number of revolutions.

During nine hours working, diagram cards and explosion records were taken every ten minutes with special indicators and samples of gas and coal in view of chemical analysis.

Mr. J. LOHMER, engineer in charge of the testing room, assisted us in the experiments and a foreman of the works managed the brake.

After one hour of preparatory running under normal conditions of the engine as well as the producer, the ashpit was emptied and cleaned and the fire scraped, and the hopper next filled up to a certain height with green coal.

A continuous test was test made at full load and at half load.

At the end of the test operations were made, as explained above, to put the generator in its original state, and after having poked the fire in all directions in the crucible so as to fill the holes which could have been formed in the fuel, fresh coal was recharged up to the initial level.

In view of controlling and insuring the accuracy of the relatively short experiments, the fuel was weighed and charged every half hour to maintain it nearly at its initial level in the hopper.

From 8.30 till 1 o'clock p.m. the engine was kept under a brake load of 149 kilogrammes (329 pounds) at the end of the lever of L = 1,648 (64″½) at an average speed of 188.66 revolutions per minute, developing:

65.11 *effective horse power.*

The corresponding consumption of brute coal was: 104.9 kilogrammes (232 pounds) during 4.30 o'clock.

Without stopping the engine, the brake load of 74 kilogrammes (163 pounds) was suddenly discharged, the load maintained being 149 − 74 = 75 kilogrammes (165 pounds).

From 1 o'clock to 5.30 o'clock, at an average speed of 195.5 revolutions per minute, the engine sustained a load of:

33.85 *effective horse power.*

The corresponding consumption of gross coal was 80 kilogrammes (176 pounds).

The two above indicated consumptions of 104.9 kilogrammes (232 pounds) and 80 kilogrammes (176 pounds) refer only to the fuel charged in the generator, without deduction of ashes, or of the 3 to 4 per cent. good coal which had been passed through the grate and picked out by sifting.

Trials of sudden total loading and unloading of the engine showed that within the space of 25 to 35 seconds the original speed was recovered according to the sensitiveness of the governor and its manner of operation.

At the other hand, 50 seconds after admission of compressed air for automatic starting, the regular speed of empty running was reached, while the governor was oscillating two or three times between its extreme positions, as shown by the ordinates representing the variations of initial pressure recorded by our "continuous explosion recorder."

At the end of the tests, all the working parts of the engine were examined, and none of them were found to have been heated.

With regard to consumption, regularity and smoothness of working, the results obtained are remarkable.

They realize a real progress in the construction of the single-acting Otto cycle engines.

With regard to the gas plant, the permanent quality of the gas produced, giving very nearly the same calorific value at both full and half load, points out the practical good proportions of the generator and the ease with which it is driven.

The table below, and the average types of diagrams and graphic records appended hereto, will confirm these statements.

EXPLANATION OF THE EXPLOSION RECORD CARD.

Taken within 90 *seconds.*

(*a*) Period of starting with compressed air.

(*b*) Period of time wanted to attain normal speed with engine running without load 50 seconds.

(*c*) First starting explosions.

(*d*) Period of oscillating of governor.

(*e*) Period of regium of explosions at empty running.

(*f*) Period of gradual loading and regium of full load.

TEST OF A 60-BRAKE HORSE POWER GAS ENGINE, TYPE G 9, WITH A SUCTION GAS PLANT OF *THE GASMOTOREN FABRIK, DEUTZ*, AT COLN, MARCH 15TH, 1904.

BY R. MATHOT, Consulting Engineer, Brussels.

TABLE OF DATA OF THE TESTS.

Diameter of Piston D = 16"5' × Piston Stroke C = 18"9'.

FULL LOAD.

1.	Average number of revolutions per minute	n.	188.66
2.	Corresponding effective work	B. H. P.	65.11
3.	Average compression, per sq. inch	lbs.	176
4.	Average initial explosive pressures, per sq. inch	,,	397
5.	" final expansion pressure, ,, ,, ,,	,,	25
6.	Vacuum at aspiration, ,, ,, ,,	,,	4.4
7.	Average mean pressure on piston, ,, ,, ,,	,,	81
8.	Corresponding indicated horse-power, per sq. inch	IHP.	77

Fuel.

9. Nature of fuel : Anthracitous coal, 10/20 m/m.
10. Origin : Coalpit of Zeche. Morsbach at Aix la Chapelle.
11. Chemical Composition of coal :

Carbon,	83.22%.	Sulphur,	0.44%.
Hydrogen,	3.31%.	Ashes,	7.33%.
Nitrogen and Oxygen,	3.01%.	Moisture,	2.69%.

12. Calorific value .... BTU. 13,650

Gas.

13. Chemical composition of gas :

Carbonic acid,	6.60%.	Marsh gas,	0.57%.
Oxygen,	0.30%.	Carbonic oxide,	24.30%.
Hydrogen,	18.90%.	Nitrogen,	49.33%.

14. Calorific value of gas, combination water at 59° Fahr., at constant volume reduced to 32° Fahr. and atm. press......... BTU. 140

Temperatures : Engine.

15. Cooling water at the inlet of the cylinder head at 55.4° Fahr. temperature at the outlet .... Deg. F. 109.5
16. Temperature at outlet of cylinder........................ ,, 127.5

Gas Generator.

17. Temperature of water in the vaporizer.................Deg. ,, 158.3

EFFICIENCIES AND CONSUMPTION.

18. Mechanical efficiency ....% 84.6
19. Consumption of brute coal per B. H. P. and per hour ....lbs. 0.85
20. Thermal efficiency related to the effective work and the brute coal consumed in the gas generator..................... ....% 24.3

HALF LOAD.

Work.

1. Average number of revolutions per minute....................n = 195.5
2. Corresponding effective work............................B. H. P. 32.83
3. ,, average compression ....lbs. 125
4. Average initial explosive pressure................... .... ,, 258
5. ,, final expansion .... ,, 18
6. Vaccuum at aspiration .... ,, 6.8
7. Average mean pressure on piston .... ,, 46.2
8. Corresponding indicated power.......................... I. H. P. 45
9. Speed variation between full and half load........................% 3.5

CONSUMPTION.

10. Consumption of brute coal per B. H. P. and per hour ....lbs. 1,155

RUNNING EMPTY.

1. Average number of revolutions per minute ....n = 199
2. Minimum corresponding compression ....lbs. 95.55
3. Average initial explosive pressure............................ ,, 220
4. ,, final expansion ,, ... ,, 0
5. Vacuum at aspiration .... ,, 8.8
6. Average mean pressure on piston .... ,, 11.2
7. Corresponding indicated horse-power ....I. H. P. 11
8. Speed variation between full load and empty running ... . % 5.2

The following report of tests made on a four cycle double-acting gas-engine of 200 horse-power in the shops of the Société des Moteurs a Gas Otto-Deutz at Cologne-Deutz is signed by Prof. Aimé Witz, Engineer of Arts and Manufactures; by Rudolph Mathot, Consulting Engineer, and by Mr. Ch. de Herbais de Thun, Engineer of Arts and Manufactures. The test was made March 14th and 15th, 1904.

Installation.

1. The motor is a single cylinder double-acting four-cycle Otto-Deutz machine, with poppet valve and electric ignition. Governing is done by the variable opening of the valve which admits the mixture. The fulcrum of one of the controlling levers is shifted by the governor. The jacket of the cylinder, the exhaust valves and the nozzle of the exhaust pipe are cooled by water circulation with visible outlet independent of each other. Each pipe is furnished with a controlling valve at its outlet. The piston and its rod are cooled by a special circulation of water with visible discharge, equally under control, and working under a pressure of about one kilogram. The pipes which bring the air and the gas have butterfly valves, enabling the quality of the mixture drawn into the cylinder to be varied.

2. An aspiration producer of the Otto-Deutz system is made up of a producer with interior vaporizer and two coke scrubbers. The producer carries at its upper part a hopper with double closure and an arrangement which prevents simultaneous operation of the two closures. The overflow from the evaporation pan can be directed either into the ash-pit or into the sewer. On leaving the producer, the gas passes down through a conduit towards the base of the first scrubber and leaves the top of the first to enter in the same manner into the base of the second, and leaves the top of the second scrubber to pass to the motor.

The following data are significant:

Diameter of the Piston	540 mm.
Stroke of Piston	700 mm.
Piston Rod Diameter	120 mm.
Diameter of Prolongation of Rod	110 mm.
Horse-Power of the Motor	200 H. P.
Revolutions per minute	150

The observations of the test covered the following points:

I. **Regularity.**
II. **Effective Power.**
III. **Consumption of Combustible.**
IV. **Consumption of Water.**
V. **Temperatures of Gas and Water.**
VI. **Analysis of Gas and Coal.**

I. *Regularity.*

Diagrams were taken and graphic registrations to inform the experimenters concerning the conditions of governing. Curves were taken when the motor was running at about half charge and simultaneously the speed of the motor was observed every thirty seconds by readings of the counter. The curves denote a great regularity in the initial pressures of the explosion. By suppressing the ignitions on one of the faces of the piston, an immediate increase in initial pressure of the explosions appeared upon the other.

By suppressing the admission of mixture, the pressure of explosions decreases gradually from 17 kilos. to 4 kilos. The mean number of revolutions was 152.22, with a range from a minimum of 144 and a maximum of 156, or a difference of 12 revolutions per minute or 8 per cent.

It is to be remarked, that between the irregular distances given on the curves between the condition for half charge and no load, the differences are proportional to the above differences of speed. The correctness of the diagram is thus confirmed. In the course of the tests on mixture, the speed of the motor was kept sensibly constant between the limits of 149.1 and 151.8 revolutions per minute. The variation is thus only 1.75 per cent. The operation of the machine was distinctly regular and there was no abnormal heating observed of any part.

II. *Effective Power.*

This was determined by means of two Prony brakes, mounted on pulleys fastened to the motor shaft. The first brake (A) has an arm of 2.04 metres, which was not exactly balanced with respect to itself. The weight necessary to balance it was 9 kilos., which should be deducted from the gross weights on the brake. The second brake (B) was exactly balanced in the presence of the observers; its lever arm was 2.06 metres. The number of turns was registered by an integrating counter mounted on cam shaft.

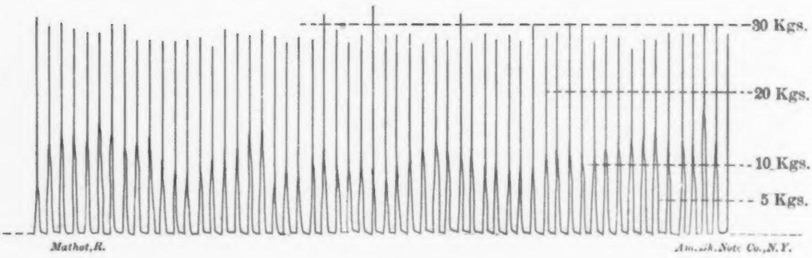


Fig. 170.

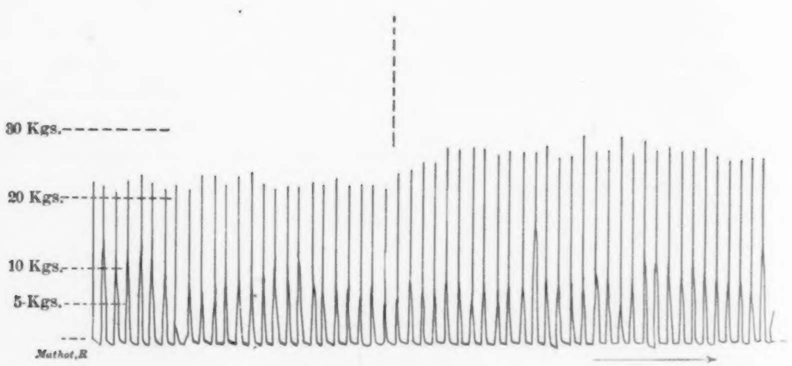


Fig. 171.

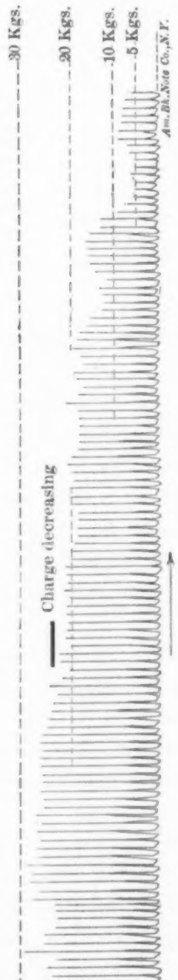


FIG. 172.

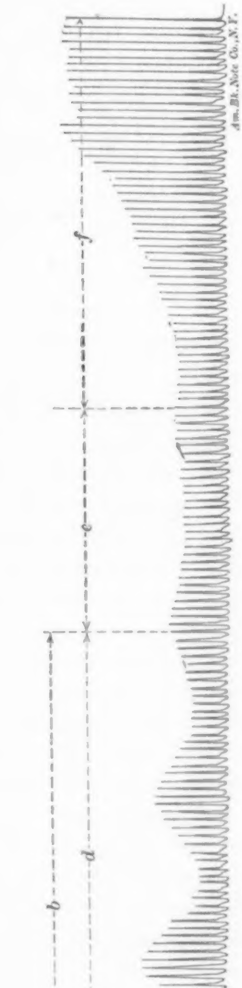


FIG. 173.

The relative readings respecting the weights on the lever arms and the speed gave the following values:

Date.	Length of Tests in Hrs.	Mean Speed. R.P.M.	Brake.	Horse Power.
Mar. 4	3	151.29	A	103.48
"	3	151.29	B	110.74
			Total:	214.22
Mar. 15	10	150.20	A	110.18
"	10	150.20	B	112.65
			Total:	222.83

Fuel Consumption.

The observations relative to fuel consumption belong to three distinct periods.

The first period, March 14, 1904, 3 P.M.—6 P.M., or three hours, covered the running of the motor and the producer being continuously charged at intervals. After stirring the fire and cleaning the grate the producer was filled with fuel to a determined level. The additions during the running were made at regular intervals and were thirty kilos. every half hour. At the end of the period, after stirring the fire and cleaning the grate, the original level was established.

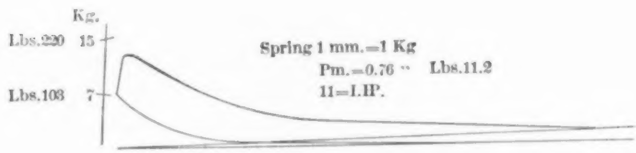
The second period, on the 14th and 15th of March, was that during which the producer was working by natural draft without addition of fuel from the end of the first period to the beginning of the third. The mechanism of the charging box was kept in a closed position during all this period by waxed seals affixed by the observers.

The third period, from 8.20 A.M. to 6.20 P.M., or ten hours, covered the regular working of the installation with continuous feeding; the operations which were performed during the first period were repeated for the third. Below are the data of the observations:

MOTOR.

Period	Length Hrs.	Mean Speed Rev. per Min.	Mean Effective Work. H.P.	Total Fuel Consmpt. Kilos.	Fuel per Hr. Kilos.	Fuel per H.P. per Hour. Grams.
I	3	151.29	214.22	270	90	420
II	14.02	..	..	21.7	1.55	..
III	10	150.20	222.83	726.7	72.67	326.12
	13		220.84	1018.4	78.34	354.73

It should be noted that at the middle and at the end of the third period there was recharged into the producer 62.5 kilos. of



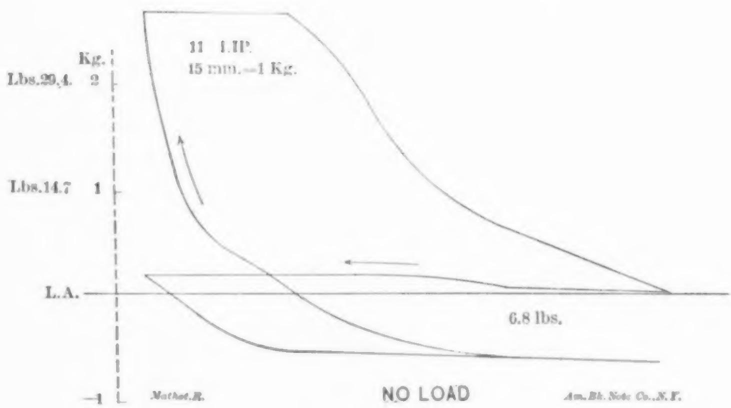


Fig. 174.

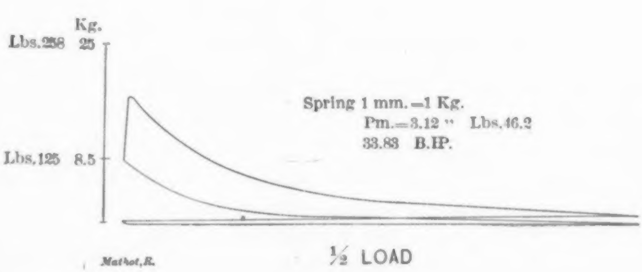


Fig. 175.

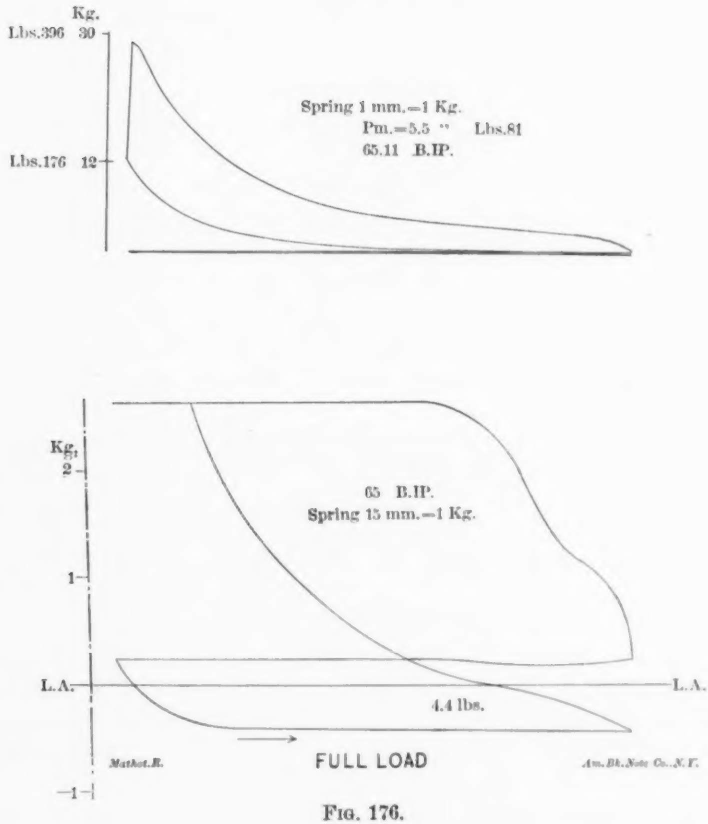

Kg.
Lbs.396 30
Spring 1 mm.=1 Kg.
Pm.=5.5 " Lbs.81
65.11 B.IP.
Lbs.176 12
Kg.
2
65 B.IP.
Spring 15 mm.=1 Kg.
1
L.A.
L.A.
4.4 lbs.
Mathot.R.
FULL LOAD
Am.Bk.Note Co., N.Y.
-1

Fig. 176.

coal cinders and utilizable small fuel which it was convenient to take note of, but of which account need not be taken.

IV. *Consumption of Water.*

Arrangements were made to secure a measurement of the necessary water both for cooling the engine itself and the cooling of the gas as well as for the generation of the steam necessary for the use of the producer. The water meters had been placed on the pipe which brings the water to the motor and on that which supplied the scrubbers.

A weir of constant output standardized by the courtesy and care of Mr. Winand to within an error of two per cent. had been established for the water leaving the piston rod. A measuring appliance was placed above the producer to measure the feed to the vaporizer, and the water escaping from the overflow of the latter was caught and weighed. The mean hourly totals are given in the following table.

The observations refer to the run in the third period of the test and correspond to the mean horse-power of 222.83 developed by the engine.

	Consumption per hour. Litres.	Mean Consumption per H.P. per hour. Litres.
For the Motor.		
Cooling of the cylinder, exhaust valves and pipe	4.650	20.870
Cooling of the piston and rod	1.750	7.853
For the producer.		
Cooling of the scrubbers	1.429	6.412
Water used in vaporizer	63.3	0.284

This last consumption referred to fuel burned gives a ratio of $\frac{286}{328.3} = 87.1$ per cent.

V. *Increase of Temperature.*

Engine.—The cooling water after its passage through the various jackets escaped to the free air through curved pipes, discharging into funnels. Each pipe was fitted with a controlling valve. The different pipes are designated as follows:

A. Exhaust pipe jacket, back end.
B. " valve jacket, back end.
C. " pipe jacket, front end.
D. " valve jacket, back end.
E. Cylinder jacket.
F. Piston and rod.

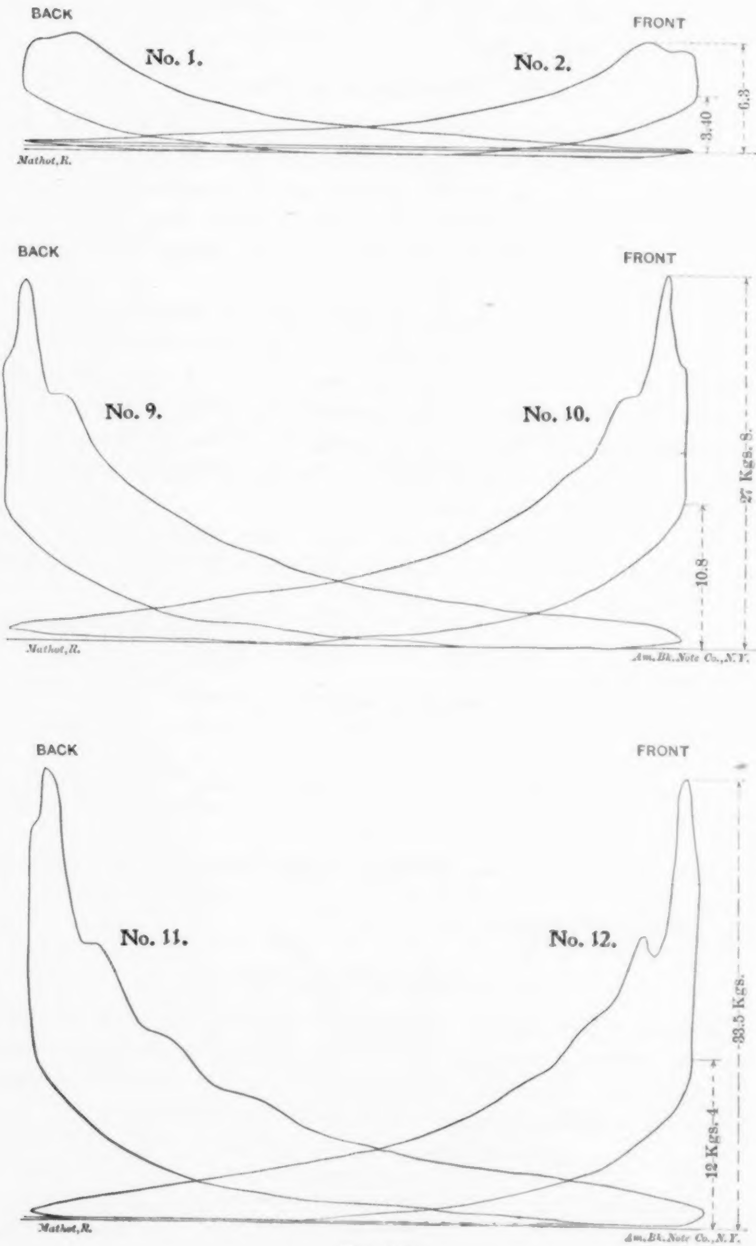

BACK
FRONT
No. 1.
No. 2.
3.40
6.3
Mathot, R.
BACK
FRONT
No. 9.
No. 10.
27 Kgs. 8.
10.8
Mathot, R.
Am. Bk. Note Co., N.Y.
BACK
FRONT
No. 11.
No. 12.
33.5 Kgs.
12 Kgs. 4
Mathot, R.
Am. Bk. Note Co., N.Y.

Fig. 177.

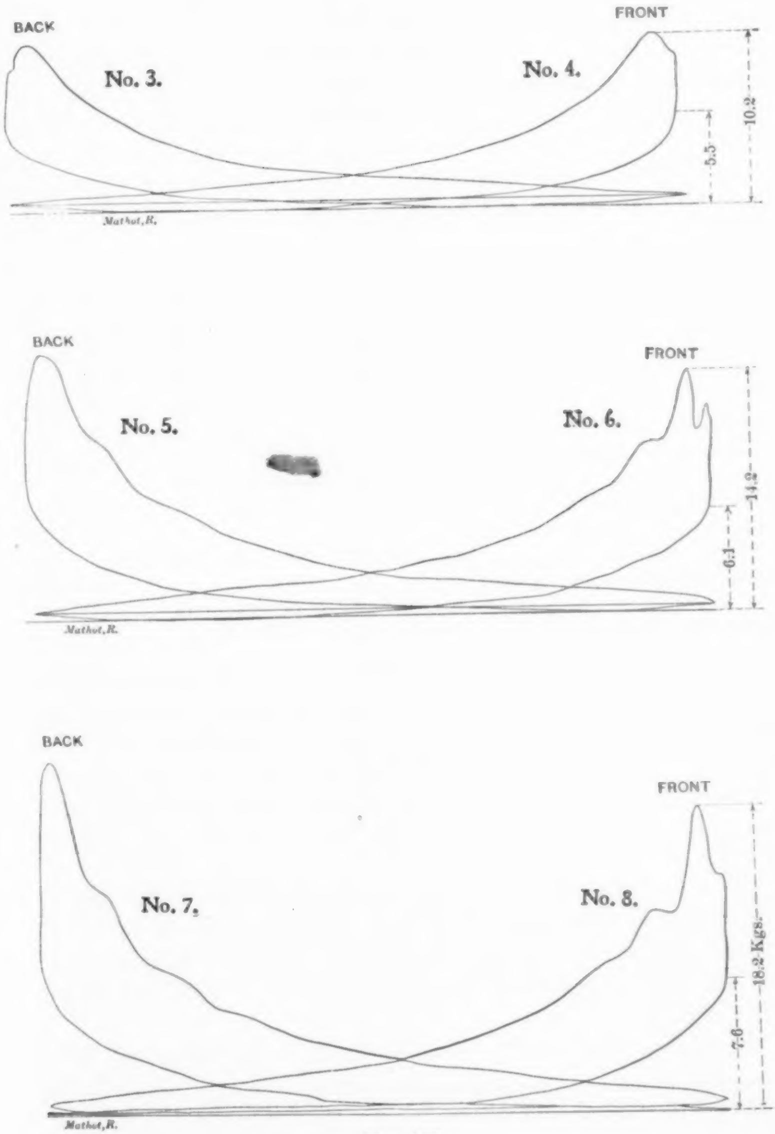

BACK
FRONT
No. 3.
No. 4.
10.2
5.5
Mathot, R.
BACK
FRONT
No. 5.
No. 6.
14.2
6.1
Mathot, R.
BACK
FRONT
No. 7.
No. 8.
18.2 Kgs.
7.6
Mathot, R.

FIG. 178.

Producer.—The water for the vaporizer had a thermometer immersed therein. The washwater for the scrubber flowed into a receiver fitted with an overflow whose contents formed the hydraulic joint. Two thermometers were placed in the gas pipe respectively at the exit from the producer and at the exit from the second scrubber. The temperatures of water and gas were taken at different intervals during the test of March 15th, and are given in the following table. The temperature of water delivered to the engine and to the scrubber remained sensibly constant at 13 degrees C.

Time.		8.40	9.10	11.30	12.40	3.15 P.M.	5.23	Mean.
Temperature of cooling water leaving the engine.	*A.*	63	50	55.5	61	74	39	57.08
	B.	51	59	57.0	63	66	64	60.08
	C.	41	61	55.5	62	62	58.5	56.60–57.18
	D.	57	65	54.0	58	64	66	60.75
	E.	40	54	60.0	65	42.5	47	51.41
Piston....		35	51	49.5	49	48.5	52.5	47.66
Water from the vaporizer of the producer.........		56	61	86.5	87	87	88	
Water leaving the scrubbers...		33	40	49	50	52.5	51	
Temperature of the gas leaving producer..................		126	178	250	280	291	294	
Temperature of gas leaving 2nd scrubber		17	19	17	18	20	19	
Temperature of the room		16	16	16	16.6	17.5	18	

From lack of preparation to this end it was not possible to measure separately the delivery from the outlets *A* and *B*, etc. We can therefore not calculate exactly the quantity of heat carried away by the circulation of water in detail, but this quantity can be approximately determined. The delivery from the pipe *E* was greater than that of each of the other pipes, but was not twice this quantity.

The mean temperature of the water leaving these five outlets will therefore be between 57.18 and 56.22, which is the mean calculated respectively on the supposition of equal discharge, and that the discharge from *C* was equal to twice that of each of the other orifices.

On this basis the heat removed per hour by the water will be:

From the cylinder and its appendages from $(57.18 - 13)\ 4650 = 205437$ calories to $(56.22 - 13)\ 4650 = 200973$ calories.

of which two-fifths or two-sixths, or 80,000 or 68,000 calories, are taken from the exhaust gases by the circulations *A* and *C*.

From the piston ....$(4766 - 13)\ 1750 = 60655$

From this is deduced that the circulation necessary to secure normal working of the engine would absorb a quantity of heat per hour approximating 180,000 to 192,000 calories, or 820 to 870 calories per horse-power per hour.

The coal used in the course of these tests was analyzed by ourselves. It contained 2 per cent. of moisture. The composition of the dryed coal was as follows:

Fixed carbon	79.8
Volatile compounds	17.1
Ash	3.1
Highest calorific power	8100 calories.

The calorific power was determined on the ground by means of the Junker calorimeter. Moreover different samples were taken whose calorific power was determined by means of the Witz eudiometric bomb and the chemical analysis was made by an engineer of the Deutz Company. The result of these analyses was as follows:

Time.	10.50	12.35	3.50 P.M.	4 35	6
CO_2			6.50%		2.90%
O			0.80%		0.50%
H			18.60%		17.70%
Methame			0.64%		0.60%
CO			24.50%		29.30%
N			48.96%		49.00%
Calorific Power from Analysis in Calories at 0°C and 760 mm highest value			1328		1446
Calorific power from the bomb and steam condensed at 15° C (highest value)	1358	1265	1287	1268	1295
Calorific power by calorimeter reduced to 0°C (highest value) 1389	1389		1335		1331
Ditto (lower value) steam condensed at 25° C	1304		1250		1246

The engine appears to us capable of developing normally without disturbance or overheating an effective horse-power of 220 horse-power and the corresponding consumption of fuel can be considered as remarkably economical. The operation of the aspirating producer is very regular, and the operations of charging and of cleansing when made regularly and at convenient time caused no disturbance in the production of gas. No deposit of tar nor of dust was detected by us and the purification of the gas appeared to be excellent. Our tests have established therefore that the double-acting four-cycle gas-engine of the Otto-Deutz Company, fed by means of an aspirating generator of the Otto-Deutz type, is capable of assuring a very regular and very economical service.

Table of the elements of the test.

ENGINE.

Piston rod diameter	540 mm
Stroke	700
Diameter of piston rod	120
" " " " prolongation	110

Full charge test.	March 14th.		March 15th
1. Mean rev. per minute	151.29		150.20
2. Effective horse-power corresponding	214.22		222.83
3. Duration of the test, hours	3		10
4. Mean temperature of water degrees, cent. after cooling piston			47.66
Same after cooling cylinder and appendages			57.18
5. Consumption of water, litres per hour for piston			1750
Producer.			
6. Fuel from Bonne Esperance et Batterie a Herstal (Belgium)			
7. Calorific power			8100 calories
8. Consumption of fuel (kilos), per hour (24-1 kilos were burned during night of 14-15)	90		72.67
9. Consumption of water litres per hour.			
Vaporizer			63.3
Scrubber			1429
10. Mean temperature of gas leaving the generator.			292°
Mean temperature of gas leaving scrubber			17°
ECONOMY.			
12. Gross consumption of coal per H. P. per hour each test-grams	420		326.12
Mean consumption including that during a stop		354.7	
Mean consumption corrected for moisture	411.16	347.61	319.60
13. Thermal economy referred to effective horse-power and dry coal consumed in producer.			
(*a*) Each test	0.190		0.244
(*b*) Mean		0.216	
14, Consumption of water per horse-power per hour in litres.			
(*a*) Cylinder and appendages			20.870
(*b*) Piston			7.853
(*c*) Vaporizer			0.284
(*d*) Washing the gas			6.412
Consumption of vaporizing water in kilos. per gram of fuel.			0.871

(Signed) *R. Mathot,*
A. Witz,
De Herbais De Thun.

*Mr. William P. Flint.**—I must plead ignorance as to the recommendation of a committee of this Society, that the total, rather than the effective British thermal units, be used for calculating gas engine efficiencies.

Where the efficiency of a gas engine is to be compared with a steam engine, the total heat is certainly the one which should be used, but where two similar gas engines are to be compared, one of which may be running, for instance, on producer gas having a large percentage of hydrogen, and the other on blast furnace gas containing practically no hydrogen, the effective British thermal units furnishes a better basis for comparison than the total does.

The method of reducing results to the arbitrary 100 horse-power basis is not alone valuable for comparing similar engines on a given gas. It has value for comparing gas engines of different types, and on different fuels, for, by eliminating the large displacement of the curves due to differences of size, it makes apparent the differences both of power obtained per unit of displacement and of economy.

It should be remembered that the basis proposed is based on the mixture volume, and not on the gas volume drawn in. The heat values of different commercial fuel gases vary widely, but the heat values of the gases per cubic foot of explosive mixture do not vary more than about 50 per cent., hence results obtained with the different gases may be intelligently compared when plotted on the lines proposed in this paper.

I wish, in closing, to thank Mr. Chambers for his interesting comments, and hope I shall have a chance to learn from him something of his experience in England.

---

* Author's closure, under the Rules.

No. 1032.*

ROAD TESTS OF CONSOLIDATION FREIGHT LOCOMOTIVES.

BY E. A. HITCHCOCK, COLUMBUS, O.
(Member of the Society.)

1. The matter contained in this paper is presented to the Society, not only for the purpose of showing the engine performance of a particular type of locomotive working under every-day road conditions, but to show by complete boiler heat balance all the losses incurred for the particular conditions encountered, and thereby being able to make comparisons of the locomotive boiler under actual working conditions, and the stationary water-tube type, as described in Paper No. 997, Volume XXIV. of the *Transactions*, the same methods for determining the heat balance being employed in all cases.

2. Through the courtesy of the officials of the Hocking Valley Railroad, and especially that of Mr. S. S. Stiffey, Superintendent of Motive Power, a series of three trials was arranged and conducted as thesis work under the direction of the writer, by Messrs. E. G. Bailey, H. E. Williams and R. E. Rightmire of the 1902 senior class in Mechanical Engineering at Ohio State University. These parties were ably assisted by Mr. W. A. Johnson of the senior class.

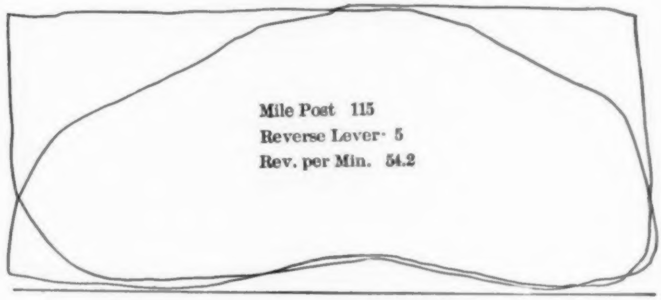
3 Also during the spring of 1903 another set of trials was made to serve as a check on the work of the previous year, slight modifications being made in connecting up some of the apparatus used and also in the application of additional apparatus.

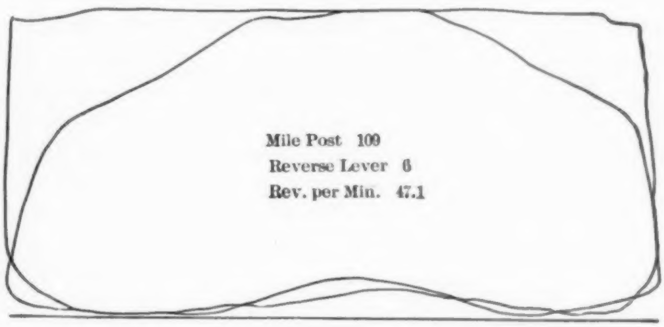
4. These trials were carried on as thesis work by Messrs. W. B. Morris, O. Z. Linxweiler, D. M. Boothman and W. R. Judson, members of the 1903 class, ably assisted by Mr. E. G. Bailey, Fellow in Experimental Engineering.

---

* Presented at the Chicago meeting (May and June, 1904) of the American Society of Mechanical Engineers, and forming part of Volume XXV. of the *Transactions*.

LEFT CYLINDER





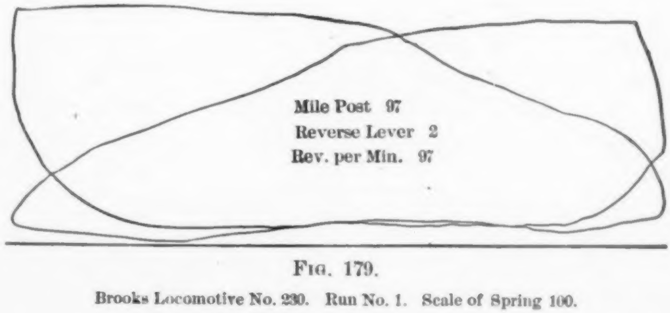


FIG. 179.

Brooks Locomotive No. 230. Run No. 1. Scale of Spring 100.

RIGHT CYLINDER

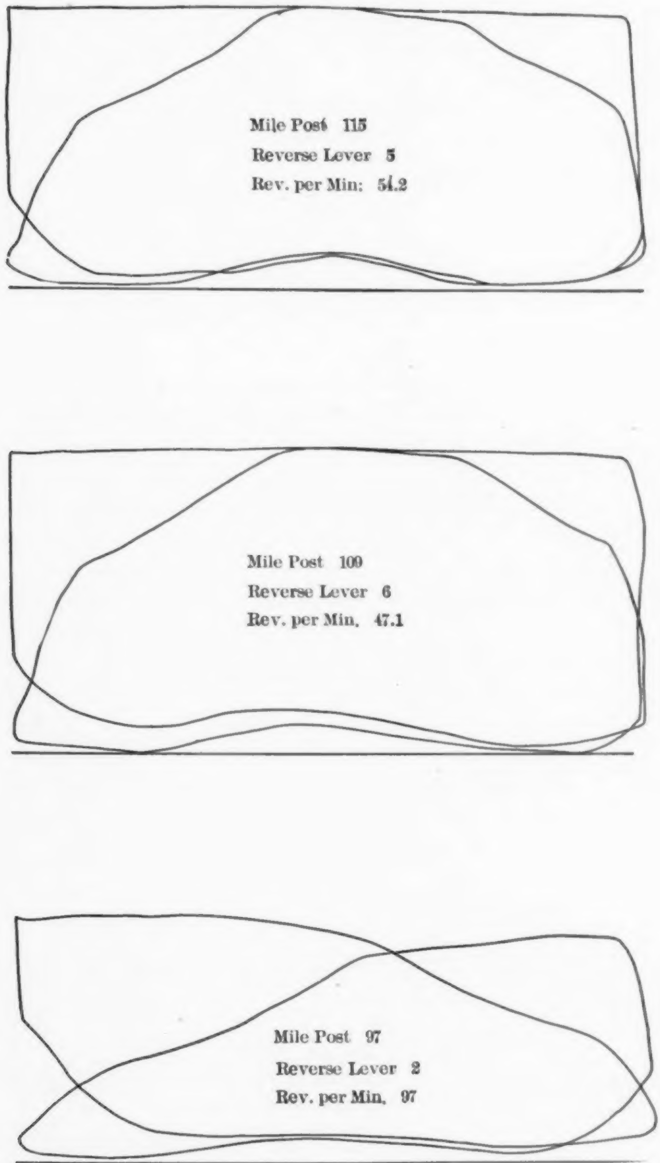


Fig. 180.

Brooks Locomotive No. 230. Run No. 1. Scale of Spring 100.

5. The locomotives used first were a Brooks and Baldwin of the ten-wheel consolidated type, dimensions for which are given in the following pages. Brooks No. 230, which had been in service about one year, was thoroughly cleaned just previous to the trial, receiving a new set of flues. Baldwin No. 240 was comparatively new, having been in service four weeks, the boiler, however, was thoroughly washed out just previous to the trial. For the trials of 1903 only one locomotive was used, Brooks No. 257, its dimensions being identical with that of No. 230. This engine was comparatively new, having been in service only four months, it simply being washed out previous to the runs.

Preparation.

6. For determination of the amount of feed water rather than use the unreliable meter, the tank of each locomotive was carefully calibrated by adding weighed quantites of water and noting the rise in the tank by means of glass tubes on scales graduated in hundreds of feet and placed on each side of the tank in the vertical plane of its center of gravity.

7. The Barrus calorimeters for moisture determinations were connected to both the steam dome and the valve chest. In the first case the sampling tube of the standard form extended alongside of the throttle valve, while for the other case the sampling tube extended into the steam chest by the way of the relief valve connection, the calorimeter connection passing through the side of the relief valve and connecting by means of an elbow to the sampling tube.

8. For obtaining temperature of escaping gases, a Hohmann and Maurer mercurial pyrometer extended through the side of the smoke box and just in front of the netting.

9. For temperature of injector discharge the branch pipes just beyond the injectors were provided with brass thermometer wells. Water manometers were located in the cab for obtaining draft in smoke box, fire box and ash pan, connection being made to these manometers by $\frac{1}{8}$-inch pipe and rubber tubing.

For obtaining samples of the products of combustion, a jacketed sampling tube was constructed of $\frac{3}{8}$-inch pipe placed inside of 1-inch pipe. Its length was 5 feet, and extended across the smoke box on the horizontal diameter just in front of the netting. The $\frac{3}{8}$-inch pipe plugged at the enclosed end had connecting to it, 6 inches apart, nipples passing through and expanded into the 1-inch

pipe. The inner end of the 1-inch jacket pipe opened into the smoke box, while to the outer end a connection was made to the discharge of the steam chest calorimeter. The sampling tube jacketed in this manner prevented any chemical reaction between the iron and CO_2 on account of the high temperature.

10. The form of apparatus used for collecting the samples of flue gas, suggested by Prof. N. W. Lord, and described and illustrated in his recent publication, "Notes on Metallurgical Analysis," p. 179, consists of a 1,000 c. c. bottle partially filled with cotton, this bottle being connected to the ⅜-inch sampling tube, and also through a tee to a two-ounce bottle containing mercury, this mercury bottle being connected to a steam aspirator receiving its steam from the steam chest. To the other branch of the tee above the two-ounce bottle was connected, by rubber tubing with a pinch cock, the tube for collecting the sample. These collecting tubes are of glass about 1 foot long, 1½ inches in diameter, drawn down at each end, and have a cubic contents of about 250 c. c. These tubes are first filled with water and closed by glass plugs put into short rubber tubes on the contracted ends. In drawing a sample of gas, a collecting tube is first connected, as spoken of above. To the lower end of this tube is connected by rubber tube a glass bulb, to the lower end of which is connected by a long rubber tube a larger bulb for retaining water when employing the collecting tube in drawing a sample of gas. After drawing a sample, and the collecting tube is free of water, it is sealed at each end with the glass plugs and then placed in an especially prepared case. This sampling apparatus was fastened to the plate on the front of the boiler, thereby making it of convenient height for the person manipulating same when standing on the pilot.

11. Crosby indicators, one on each cylinder, with three-way cocks were used. The connecting pipes were ¾-inch with easy bends and well covered with magnesia. Scale of spring 100.

12. The reducing motion was the double-slotted pendulum type with sliding block connected to cross head, and also a sliding block receiving motion for the indicator and transmitting such to same by a rigid connection, so that the indicator cord used was not over 1 foot long.

13. For obtaining the number of revolutions a continuous counter was fastened to one of the forward boiler braces in a convenient position to read, receiving its motion from the reducing motion.

LEFT CYLINDER

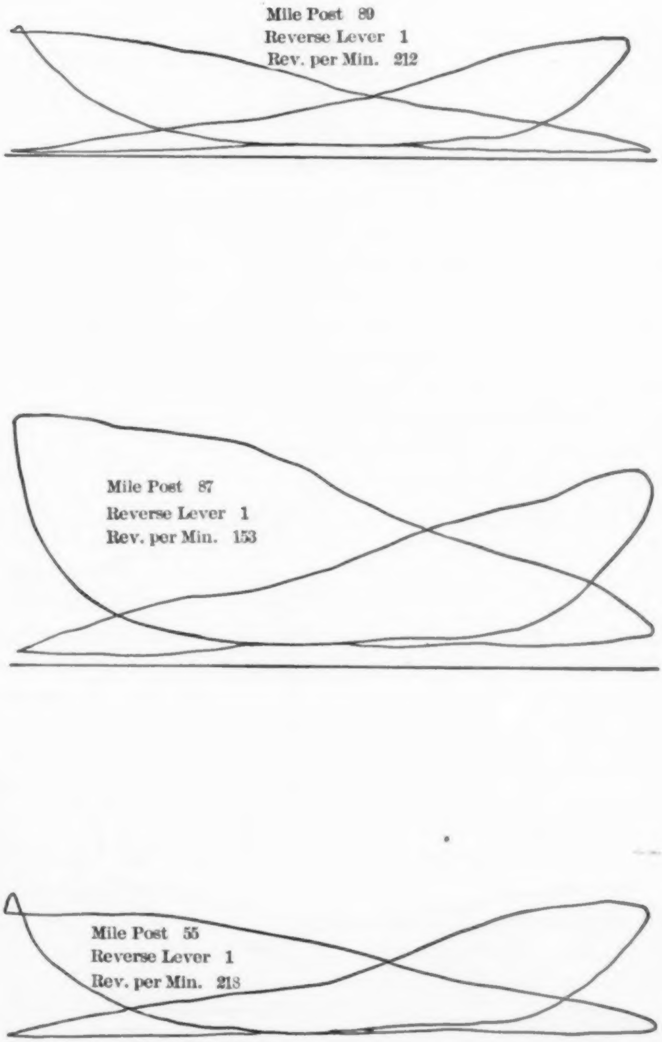


FIG. 181.

Brooks Locomotive No. 230. Run No. 1. Scale of Spring 100.

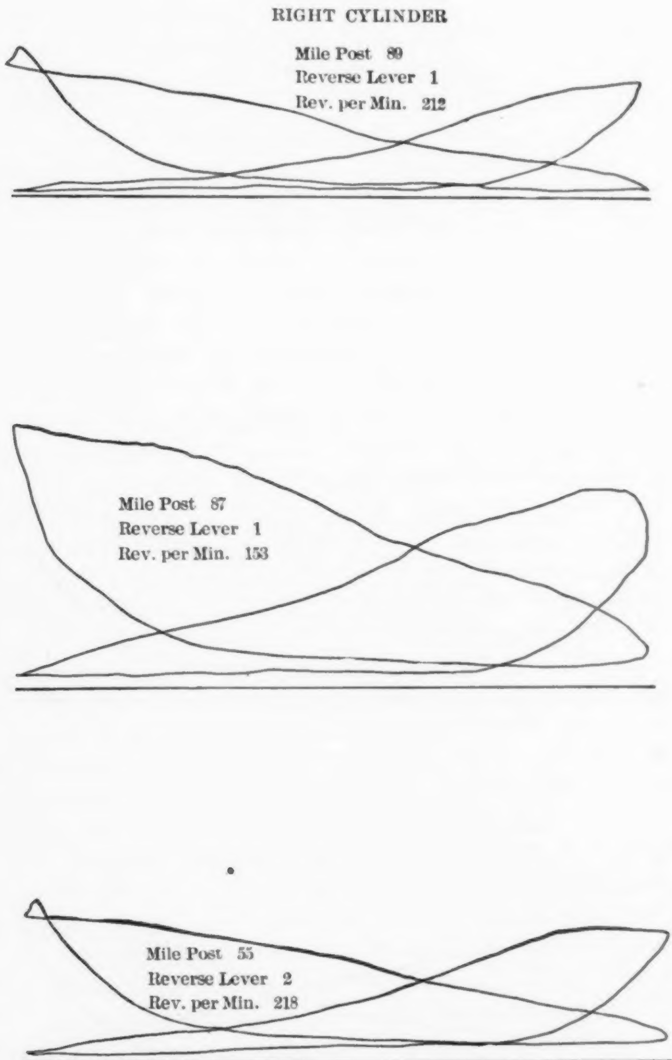


FIG. 182.

Brooks Locomotive No. 230. Run No. 1. Scale of Spring 100.

14. A continuous counter was also connected to the air pump, receiving its motion from a clamp on the piston rod, the rod slipping through the clamp when against either stuffing box.

Method of Conducting Trials.

15. The coal was weighed in small quantities and thrown on the tender loose, with the exception of from 1,500 to 2,000 pounds, which was placed in sacks of 100 pounds each, to be used at the end of the trial. At the time of coaling a sample was taken for analysis.

16. The fire was cleaned about one and one-half hours before starting. Just before coupling onto the train the fire was shaken down and ash pan cleaned, when at the start the depth of the fire was noted and also height of water in the boiler, or in other words, the alternate method for starting.

At the end of the trials the same method was pursued and refuse drawn from the pan, weighed and sampled for analysis.

17. The number of injector applications was noted and a corresponding correction made for the loss at the overflow, also the number of times of popping of safety valve with duration, the loss of steam from this direction being determined from a test of the pop valve.

18. All readings and indicator cards were taken at every mile post over the heaviest grade of the road, but when on the lighter or reverse grades, on account of the much higher rate of speed, readings were taken at every second mile post. The time at passing mile posts was noted and also the time by stop watch that the revolution counter indicated 30 revolutions, from which revolutions per minute were obtained.

19. The flue gas samples for the whole trip were taken at every fourth mile post and these samples analyzed by the Orsat apparatus the following day.

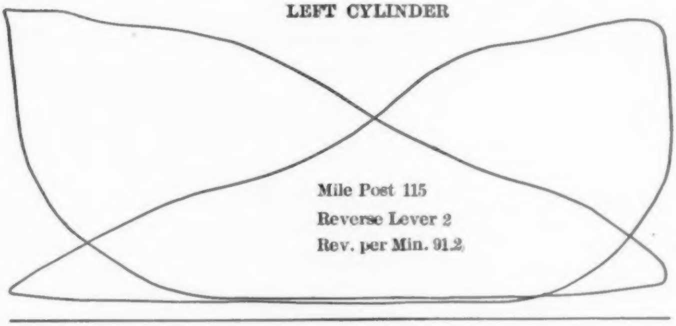
20. In making the first three trials, it was anticipated that sufficient cinder and combustible matter for analysis would collect in the front end, but as the two engines thoroughly cleaned themselves, it was necessary to make a subsequent run under similar conditions, catching an average sample of combustible matter leaving the stack. This was done by a spark collector fastened to the rear of the stack. This collector was a tight pine box 3 feet long, 2 feet deep and 1 foot wide. A funnelshaped piece of galvanized iron pipe, 4½ inches diameter at the large end,

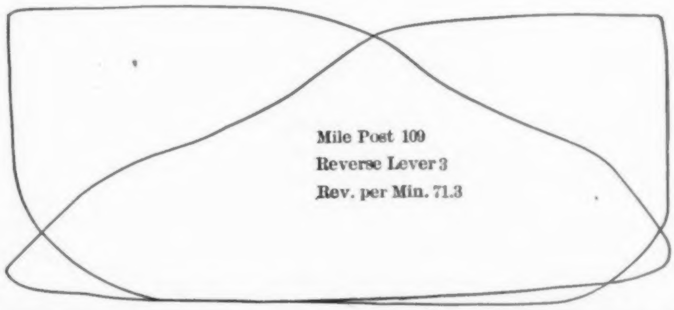
passed through the lid of the box at one end, reaching within 9 inches of the bottom. To the upper or small end of this conical pipe, was connected a ½-inch "return bend" brass pipe with chamferred end, extending just down into the stack. This brass pipe was free to swing across the stack at an angle of 45 degrees to the center line of the double exhaust nozzle, the movement being controlled from the running board and so arranged as to lock in three different positions. The interior of the box was divided by two baffle plates, so as to change direction of gases passing through the box, thereby making deposit of solid matter more positive, and in order to relieve the pressure caused by the exhaust at the same time preventing escape of combustible matter, about 30 per cent. of the lid at the opposite end from the conical pipe, was made of fine wire gauze and so arranged that this gauze could be easily removed and cleaned, as the smoke in combination with the steam would after a time clog the screen. From the stack sample and the analysis of the coal and refuse on this auxiliary trial, the loss through the stack for the previous trials was computed.

21. On the four trials of the following year this same apparatus was used, but improved somewhat by lining the box with galvanized iron, as it was found that the lumber soon dried out and opened up sufficiently in places to admit exit of soot. This was entirely eliminated with the lining, the only opportunity for escape of the products of combustion being through the brass gauze.

22. The four trials of 1903 were conducted in the same manner as the three of the previous year with some slight additions of apparatus, change in location, etc., *i.e.*, a Thompson indicator was connected to the steam chest on one side while, as before, a Barrus calorimeter connected with the opposite side. These were reversed on trials No. 6 and No. 7. The steam aspirator for the flue gas sampling apparatus was connected to the calorimeter sampling tube at the dome, as was also the steam gauge. The aspirator located at this point with ⅛-inch pipe leading to the front end gave much better satisfaction since the aspiration was constant and not intermittent when connected to the steam chest as in trials No. 1 and No. 3. The pressure gauge connected to the dome had its advantages in location over the regular gauge in the cab, since it could be a standardized gauge, easily removed between runs and standardized at any time.

LEFT CYLINDER





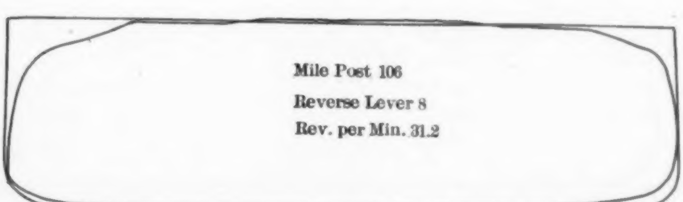


Fig. 183.

Baldwin Locomotive No. 240. Run No. 3. Scale of Spring 100.

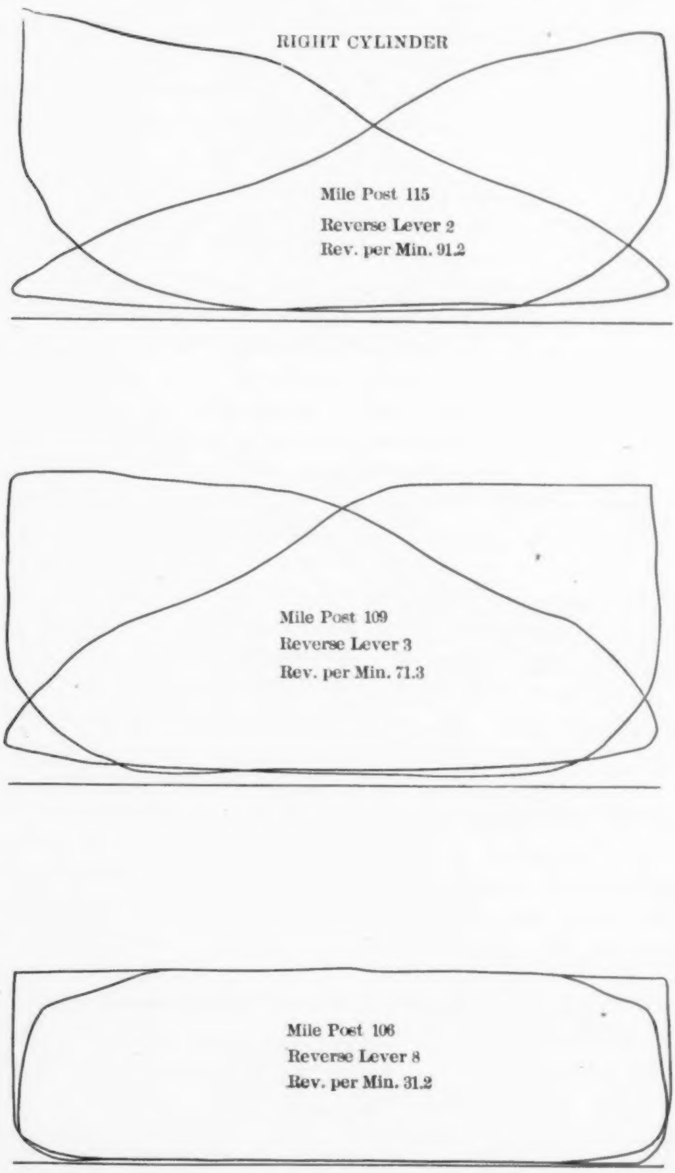


FIG. 184.

Baldwin Locomotive No. 240. Run No. 3. Scale of Spring 100.

23. The airpump counter was placed in a more convenient position, *i.e.*, on top of the air cylinder and worked by means of a small plunger working in a brass cylinder connected to the air pump. The amount of steam used by the air pump in all cases was computed from data furnished by the New York Air Brake Co.

24. For assistance rendered and suggestions given we are indebted to Prof. N. W. Lord, in whose department all the chemical work was carried on and calorific determinations of coal and stack refuse made.

General Dimensions.

H. V. Railway Locomotives.	No. 230.	No. 240.
Make	Brooks.	Baldwin.
Weight on drivers	133,500 lbs.	131,090 lbs.
Weight on truck wheels	17,000 lbs.	14,650 lbs.
Total weight	150,500 lbs.	145,740 lbs.
Weight of tender loaded	104,000 lbs.	100,000 lbs.
Wheel base, driving	15 feet.	15 feet.
" " total of engine	23 ft. 1 in.	23 feet.
" " total of engine and tender	....	....
Total length, engine	37 ft. 9 ins.	37 ft. 9 ins.
" " engine and tender	61 ft. 0 in.	61 ft. 0 in.
Height, center of boiler above rails	8 ft. 4 ins.	7 ft. 11¼ ins.
" top of stack above rails	14 ft. 5½ ins.	12 ft. 5¾ ins.
Heating surface, fire-box	147.7 sq. ft.	158.9 sq. ft.
" " tubes	1,732.8 sq. ft.	1,727.8 sq. ft.
" " total	1,880.5 sq. ft.	1,886.7 sq. ft.
Grate area	31.08 sq. ft.	30.1 sq. ft.
Drivers, diameter	54 ins.	54 ins.
" material of centers	steeled c. i.	steeled c. i.
Main wheel fit	7⅞ ins.	8 ins.
Truck wheels, diameter	30 ins.	30 ins.
Journals, driving axles, diameter	8 ins.	8½ ins. & 8 ins.
" truck axle, diameter	5 ins.	5 ins.
Main crank-pin, diameter	6¼ ins.	5¾ ins.
" " length	6¼ ins.	5½ ins.
Cylinder diameter	20 ins.	20 ins.
Piston stroke	26 ins.	26 ins.
Piston rod diameter	3¾ ins.	3½ ins.
Valve rod diameter	1¾ ins.	1½ ins.
Kind of piston rod packing	metallic.	metallic.
Length of main rod, center to center	10 ft. 7½ ins.	10 ft. 2⅝ ins.
Steam ports, length	18½ ins.	16 ins.
" " width	1⅝ ins.	1¼ ins.
Exhaust ports, length	18½ ins.	16 ins.
" " width	3 ins.	2½ ins.
Bridge, width	1½ ins.	1 in.

H. V. Railway Locomotives.	No. 230.	No. 240.
Valves, kind	balanced slide.	balanced slide.
" greatest travel	5½ ins.	5½ ins.
" outside lap	1 in.	¾ in.
" inside lap	0	0
Boiler, type	Belpaire.	Belpaire.
Working steam pressure	180 lbs.	180 lbs.
Material in barrel	carbon steel.	carbon steel.
Thickness of material	$\frac{9}{16}$ in.	$\frac{9}{16}$ in. and ⅝ in.
Barrel, outside diameters	63⅝ ins. & 60 ins.	60 ins. & 66½ ins.
Kind of seams, longitudinal	sextuple butt and quintuple lap.	butt joint—double riveted.
" " " circumferential	double lap	lap joint, double riveted.
Tube sheet, thickness	½ in.	½ in.
Crown sheet, thickness	....	⅜ in.
Kind of crown sheet stays	radial.	radial.
Dome, diameter outside	31 ins.	31 ins.
Firebox, length	9 ft.	9 ft.
" width	3 ft. 5 ins.	3 ft. 4 ins.
" depth, front	5 ft.	4 ft. 9½ ins.
" " back	4 ft. 7 ins.	4 ft. 9½ ins.
Distance of grate below fire door	27 ins.	18½ ins.
Firebox, material	carbon steel.	carbon steel.
" thickness of sheets	½ in.	⅜ in.
" width of water space	4 ins.	4 ins.
Grate, length	9 ft.	9 ft.
" width	3 ft. 5 ins.	3 ft. 4 ins.
" kind	rocking	rocking
" number of bars	11	11
Tubes, number	241	240
" material	Shelby drawn steel.	cold drawn steel
" outside diameter	2 ins.	2 ins.
" length over sheets	13 ft. 10⅛ ins.	13 ft. 10 ins.
Smoke box, inside diameter	62 ins.	60 ins.
" " length	53 ins.	62 ins.
Exhaust nozzle, single or double	single.	double.
" " fixed or variable	fixed.	fixed.
" " diameter	4⅛ ins.	3¾ ins.
Distance of tip from center of boiler	4 ins. above.	0
Netting, kind	stamped.	stamped.
" size of mesh	¼ in. x 1¼ ins.	¼ in. x 1½ ins.
Stack, kind	straight.	straight.
" diameter	15½ ins.	17½ ins.
" height above smoke box	42 ins.	....
Ratio of air space to grate area	47 per cent.	36 per cent.
Width of air space	⅞ in.	⅝ in.
Tender, type	8-wheeled.	8-wheeled.
Tank capacity	5,240 gals.	4,815 gals.
Kind of material	¼ in. steel.	¼ in. steel.
Wheels, diameter	33 ins.	33 ins.

LEFT CYLINDER

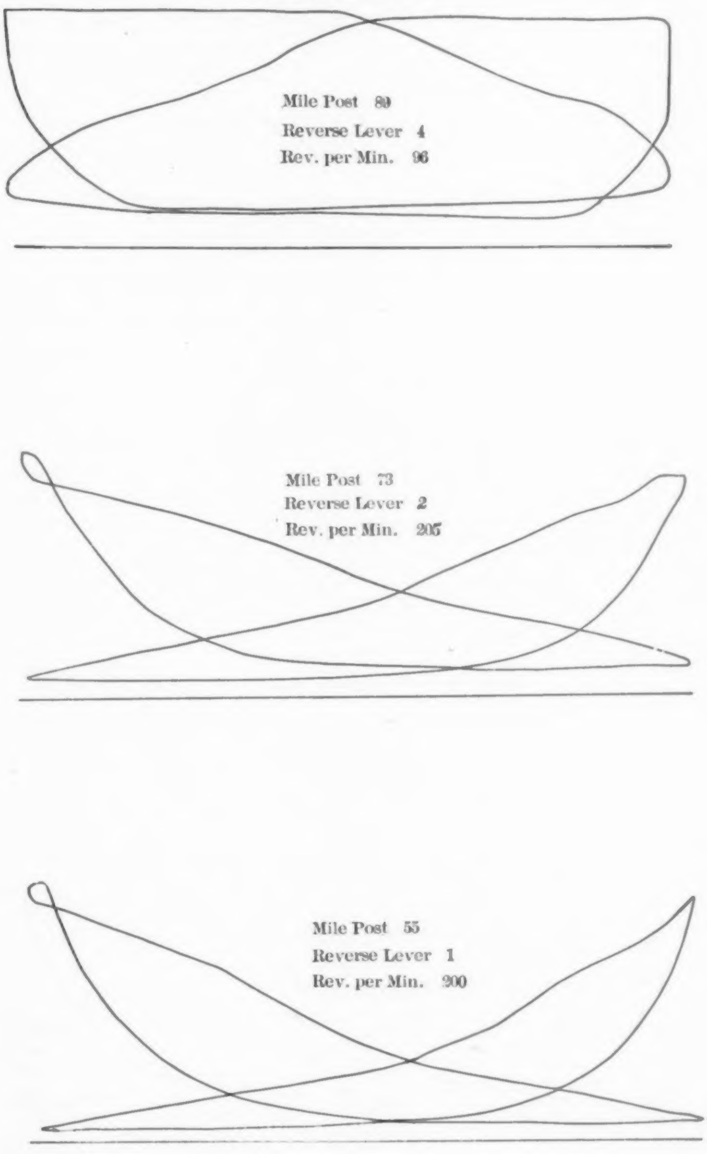


Fig. 185.

Baldwin Locomotive No. 240. Run No. 3. Scale of Spring 100.

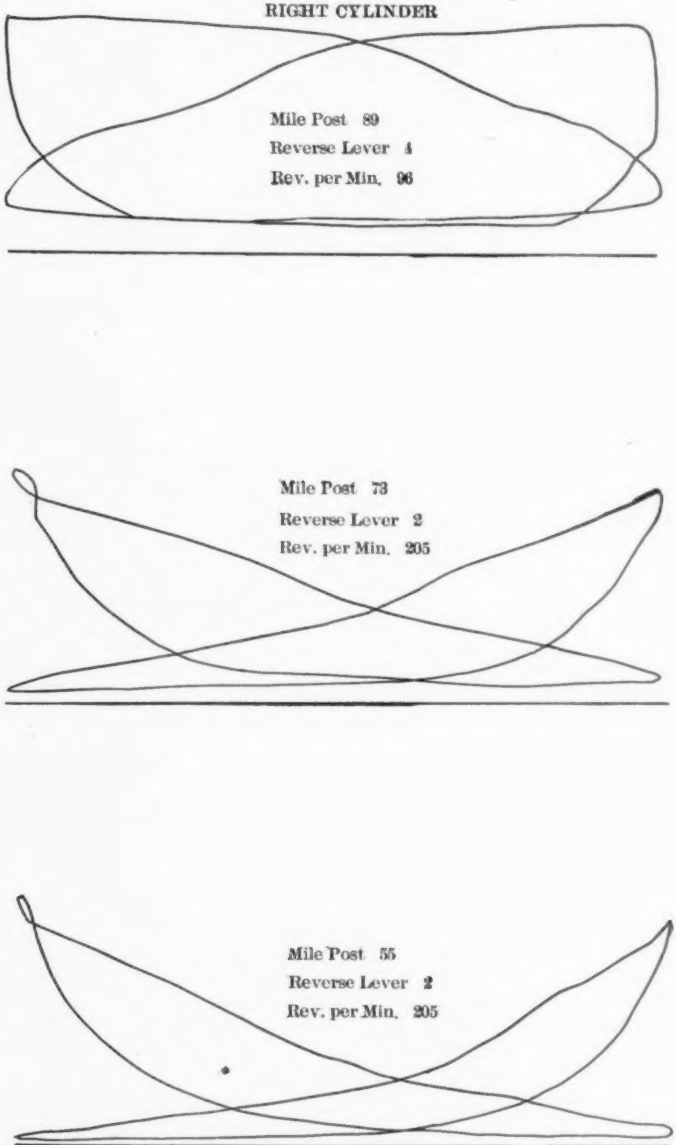


FIG. 186.

Baldwin Locomotive No. 240. Run No. 3. Scale of Spring 100.

H. V. Railway Locomotives.	No. 230.	No. 240.
Journals, diameter	4½ ins.	....
" length	8 ins.	....
Distance between centers	5 ft.	5 ft. 4 ins.
Wheel fit	....	5⅝ ins.
Diameter of center of axle	4⅝ ins.	....
Length of tender over bumper bar	23 ft. 3 ins.	23 ft 3 ins.
Length of tank	19 ft. 6 ins.	21 ft. 7 ins.
Width of tank	9 ft. 11 ins.	9 ft. 3½ ins.
Height of tank to collar	4 ft. 6 in.	4 ft. 10 ins.
Tires, make	midvale, open hearth.	st'd steel works.
Sight feed lubricator	Michigan.	Michigan.
Front and back couplers	Buckeye.	Buckeye.
Safety valves	Coale.	Coale 3 ins muffled.
Sanding devices	Leach.	Leach "D" single.
Injectors	Ohio.	Ohio.
Air brake equipment	New York.	New York.
Tender brake beam	Marden.	Marden.
Tender brake shoe	Brooks.	Marden.
Steam gauges	Brooks steeled.	Ashcroft.

1. Number of trial	1	3
2. Date of trials	May 9.	May 20, 1902
3. Duration of trials—total, hours	6.23	5.49
4. Duration of trials, running time	4.65	4.47
5. Duration of trials, steaming time	4.51	4.03
6. Number of stops	5	5
7. Kind of fuel	Hocking.	Hocking.
8. State of weather	Clear.	Cloudy & clear.
9. Direction of wind	N. E.	W. by S. W.
10. Velocity of wind, miles per hour	11.5	5

Average Pressures.

11. Steam pressure at dome	lbs.	166.4	170.1
12. Barometer	ins.	30.2	30.
13. Absolute steam pressure	lbs.	181.2	184.8
14. Force of draft, front end	ins.	3.78	4.12
15. Force of draft, fire box	ins.	1.75	2.16
16. Force of draft, ash pan	ins.	.19	.88

Average Temperatures.

17. External air	deg. Fahr.	55.6	83.
18. Escaping gases	" "	745.3	730.
19. Feed water in tank	" "	63.5	72.4
20. Feed water leaving injector	" "	194.	195.

Fuel.

21. Size		Lump.	Lump.
22. Thickness of fire	ins.	10 to 12 ins.	12 to 14 ins
23. Weight of coal fired	lbs.	13,240	10,266

H. V. Railway Locomotives.		No. 230.	No. 240.
24. Percentage of moisture in coal by analysis.	p. c.	6.72	6.86
25. Weight of dry coal fired	lbs.	12,351	9,561
26. Weight of refuse in pan	lbs.	580	398
27. Percentage of refuse in pan to coal	p. c.	4.38	3.88
28. Combustible in refuse	lbs.	134	102
29. Total combustible minus comb. in pan	lbs.	10,985	8,687
30. Totol ash by analysis	lbs.	1,231	772
31. Weight of ash passing flues	lbs.	785	476
32. Total refuse passing flues	lbs.	2,643	1,505
33. Percentage of ash lost to total ash	p. c.	63.85	61.66
34. Percentage of refuse through stack to coal	p. c.	19.96	14.66
35. Equivalent coal actually burned	lbs.	10,885.	8,927.
36. Net dry coal burned	lbs.	10,154	[illegible],315
37. Net combustible burned	lbs.	9,127	7,658.

Fuel Analysis—Proximate Analysis.

38. Fixed carbon	per cent.	48.69	51.25
39. Volatile matter	" "	35.29	34.38
40. Moisture	" "	6.72	6 86
41. Ash	" "	9.30	7.51

Ultimate Analysis.

42. Carbon	per cent.	65.59	68.12
43. Hydrogen	" "	5.15	5.34
44. Oxygen	" "	16.43	16.96
45. Nitrogen	" "	1.39	1.44
46. Sulphur	" "	2.13	.63
47. Ash	" "	9.30	7.51

Analysis of Pan Refuse.

48. Combustible	per cent.	23.05	25.65
49. Ash	" "	76.95	74.35

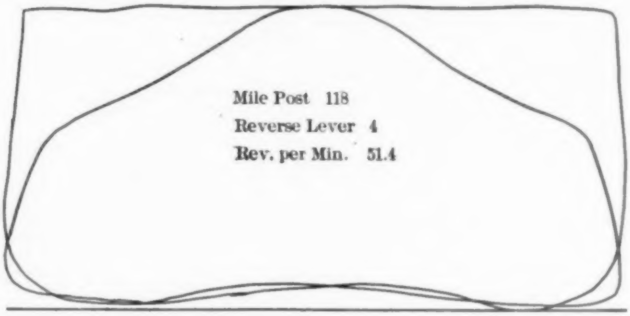
Analysis of Stack Refuse.

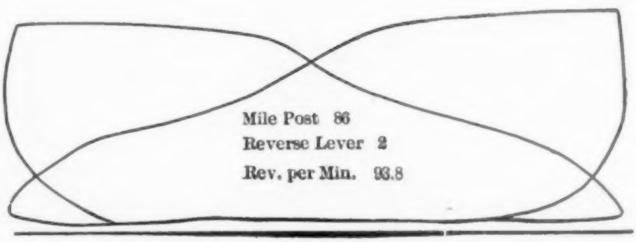
50. Combustible	per cent.	70.30	68.37
51. Ash	" "	29.70	31.63

Fuel per Hour Steaming Time.

52. Actual coal fired	lbs.	2,936.	2,547.
53. Equivalent coal burned	lbs.	2,413.	2,215.
54. Combustible burned	lbs.	2,024.	1,899.
55. Actual coal fired per square foot grate	lbs.	94.46	84 62
56. Equivalent coal burned per square foot grate		77.64	73.60
57. Combustible burned per square foot grate		65.12	63.10
58. Combustible burned per square foot heating surface		1.076	1.006

LEFT CYLINDER





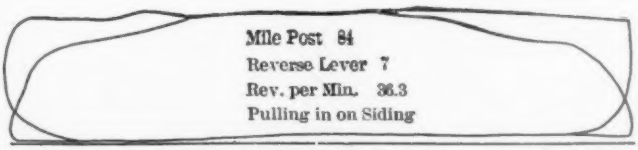


FIG. 187.

Brooks Locomotive No. 257. Run No. 4. Scale of Spring 100.

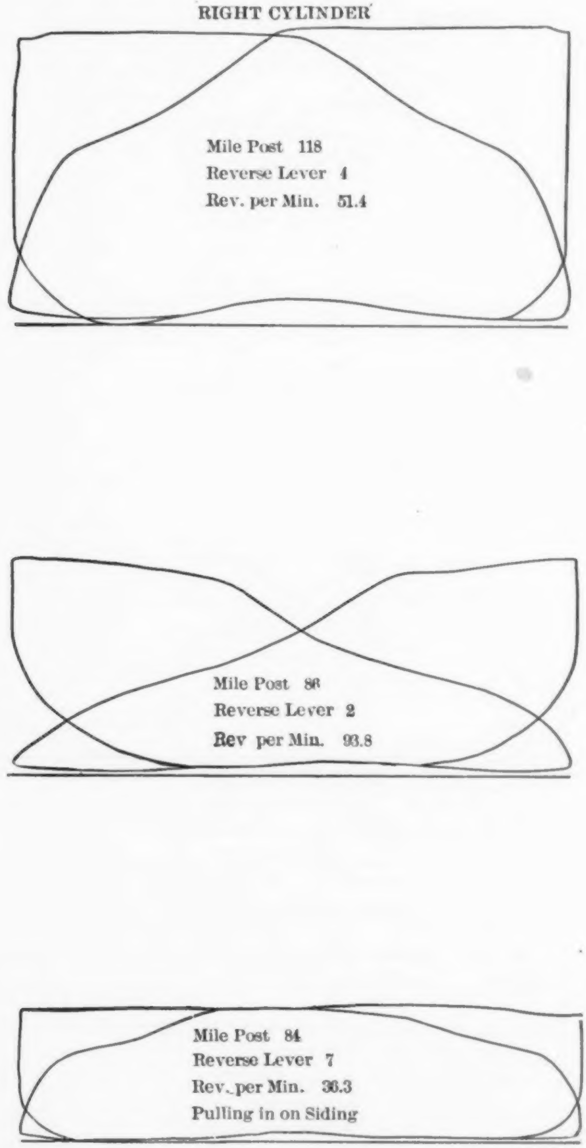


FIG. 188.

Brooks Locomotive No. 257. Run No. 4. Scale of Spring 100.

H. V. Railway Locomotives.	No. 230.	No. 240.
Calorific Value of Fuel.		
59. Calorific value per pound of actual coal by Mahler Calorimeter	11,898	12,265
60. Calorific value per pound of dry coal	12,755	13,168
61. Calorific value per pound of combustible	14,167	14,323
62. Calorific value per pound of stack refuse	9,983	9,863
Quality of Steam.		
63. Quality of steam at dome (dry steam = unity)	.9609	.9632
64. Quality of steam at steam chest	.9853	.9621
65. Quality correction (steam at dome)	.9715	.9731
Water.		
66.* Actual weight of water fed to boiler ... lbs.	82,710	71,240
67. Equivalent weight of water actually evaporated into dry steam	80,355	69,323
68. Factor of evaporation	1.2056	1.1969
69. Equivalent water evaporated into dry steam from and at 212 degrees	96,876	82,973
Water per Hour Steaming Time.		
70. Equivalent evaporation per hour from and at 212 deg.	21,480	20,570
71. Equivalent evaporation per hour from and at 212 degrees per square foot heating surface	11.42	10.91
72. Horse-power developed by boiler (34.5 lb. rating)	622.	596.
Economic Results.		
73. Water apparently evaporated under actual conditions per pound of coal as fired	6.25	6.94
74. Equivalent evaporation from and at 212 degrees per pound of actual coal fired	7.32	8.08
75. Equivalent evaporation from and at 212 degrees per pound of dry coal fired	7.84	8.68
76. Equivalent evaporation from and at 212 degrees per pound of combustible minus pan combustible	8.82	9.55
77. Equivalent evaporation from and at 212 degrees per pound of equivalent coal burned	8.90	9.29
78. Equivalent evaporation from and at 212 degrees per pound of dry coal burned	9.54	9.98
79. Equivalent evaporation from and at 212 degrees per pound of combustible burned	10.61	10.84
Efficiencies.		
80. Efficiency of boiler from combustible minus pan combustible	60.12	64.38
81. Efficiency of boiler from combustible burned	73.03	73.16
82. Efficiency of boiler and furnace from coal fired	59.39	63.65

* Corrected for height of water in boiler and overflow from injector.

Flue Gas Analysis by Weight.

H. V. Railway Locomotives.	No. 230.	No. 240.
83. CO_2 = Carbon dioxide	17.72	13.87
84. O = Oxygen	6.44	10.31
85. CO = Carbon monoxide	.64	.02
86. N = Nitrogen	75.20	75.80
87. Percentage of air excess	39.9	82 9

Heat Balance.

	B. T. U.	p. c.	B. T. U.	p. c.
88. Loss per lb. of coal due to products of combustion	1,355	11.39	1,252	10.21
89. Loss per lb. of coal due to air excess	448	3.77	875	7.13
90. Loss per lb. of coal due to latent heat	502	4.22	504	4.11
91. Loss per lb. of coal due to unburned coal in pan	132	1.11	132	1.08
92. Loss per lb. of coal due to unburned coal out stack	1,992	16.73	1,466	12 95
93. Loss per lb. of coal due to CO	292	2.46	.....	.....
94. Loss per lb. of coal due to radiation, etc.	111	.93	230	1.87
95. Heat used in evaporation	7,066	59.39	7,806	63.65
96. Total heat supplied	11,898	100.00	12,265	100.00

Engine.

Number of cars in train	35	34
Gross load in tons	1,669	1,677
Total miles	73.91	73.91
Total revolutions	......	27,910
Average speed in miles per hour	15.91	16.53
Average revolutions per minute	99.1	103.
Steam pressure by gauge	166.4	170.1
Initial pressures, right cylinder head end	111.6	145.3
" " " " crank end	100.5	136.9
" " left " head end	103.9	147.9
" " " " crank end	103.	141.1
Back pressure, right cylinder head end	10.9	11.9
" " " " crank end	6.7	10.8
" " left " head end	8.7	14 4
" " " " crank end	7.7	10.8
Mean effective pressure, right cylinder head end	71.7	63.8
" " " " " crank end	62.9	57.5
" " " left " head end	77.0	64.6
" " " " " crank end	65.2	59.7
Horse-power, right cylinder head end	145.6	159.8
" " " crank end	117.5	135.4
" left " head end	155.	159.7
" " " crank end	120.3	141.7
Total	538.4	596.6
Exhaust horse-power	71.3	79.6

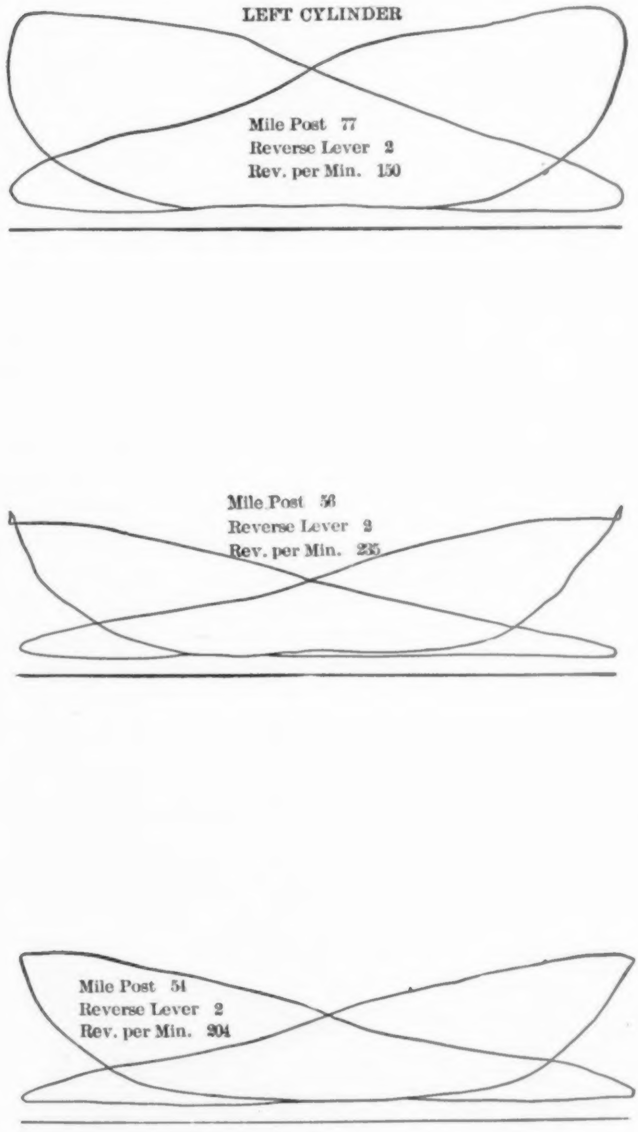


FIG. 189.

Brooks Locomotive No. 257. Run No. 4. Scale of Spring 100.

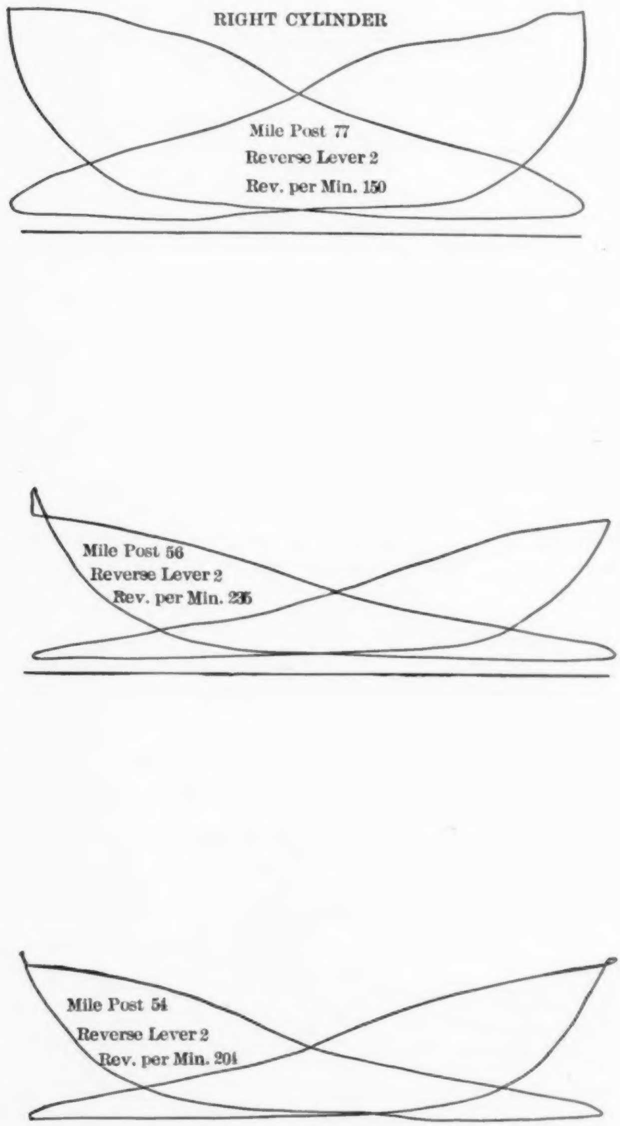


FIG. 190.

Brooks Locomotive No. 257. Run No. 4. Scale of Spring 100.

H. V. Railway Locomotives.	No. 257.	No. 257.
Safety valve, number of times popped	13	6
" " number of seconds on	2,110	840
" " pounds of steam escaped	1,114	443
Steam lost at calorimeters, dome	336	327
" " " " valve chest	195	233
Dry steam lost by pop and calorimeters	1,610	966
Dry steam used by engine and air pump	78,745	68,357
Total hours throttle open	4.51	4.03
Indicated horse-power, hours	2,428	2,404
Pounds dry steam per indicated horse-power per hour including air pump	32.47	28.42
Total strokes of air pump	5,878	5,173
Dry steam used by air pump ... lbs.	855	754
Pounds dry steam per indicated horse-power per engine alone.	32.07	28.11
Ton miles	123,351	123,947
Pounds coal per ton mile ... lbs.	.1074	.0828
Pounds steam per ton mile ... lbs.	.638	.549

1. Number of trial	4	5
2. Date of trials	May 4.	May 6, 1903.
3. Duration of trials ... total hrs.	7.12	5.81
4. Duration of trials ... running time.	4.91	4.72
5. Duration of trials ... steaming time.	4.53	4.00
6. Number of stops	8	7
7. Kind of fuel	West Va.	West Va.
8. State of weather	Clear.	Clear.
9. Direction of wind	S. E.	S. W.
10. Velocity of wind ... miles per hr.	6.2	7.7

Average Pressures.

11. Steam pressure at dome ... lbs.	180.5	167.
12. Barometer ... ins.	29.5	29.36
13. Absolute steam pressure ... lbs.	194.9	181.5
14. Force of draft—front end ... ins.	3.86	4.2
15. Force of draft—fire box ... "	1.74	1.83
16. Force of draft—ash pan ... "	.19	.13

Average Temperature.

17. External air ... deg. Fahr.	59.	71.
18. Escaping gases ... "	760.4	742.6
19. Feed-water in tank ... "	55.4	55.8
20. Feed-water leaving injector ... "	204.	198.

Fuel.

21. Size	Lump.	Lump.
22. Thickness of fire ... ins.	8 to 12	8 to 10
23. Weight of coal fired ... lbs.	10,730	9,800
24. Percentage of moisture in coal by analysis, per cent.	2.85	2.20

H. V. Railway Locomotives.		No. 257.	No. 257.
25. Weight of dry coal fired	lbs.	10,424	9,584
26. Weight of refuse in pan	"	429	539
27. Percentage of refuse in pan to coal	per cent.	4.	5.5
28. Combustible in refuse	lbs.	119.	188.
29. Total combustible minus comb. in pan	"	9,425.	8,552.
30. Total ash by analysis	"	880.	834.
31. Weight of ash passing flues	"	570.	485.
32. Total refuse passing flues	"	2,033.	1,516.
33. Percentage of ash lost to total ash	per cent.	64.8	57.9
34. Percentage of refuse through stack to coal.	" "	18.94	15.48
35. Equivalent coal actually burned	lbs.	9,081.	8,537.
36. Net dry coal burned	"	8,822.	8,350.
37. Net combustible burned	"	8,078.	7,623.

Fuel Analysis.—Proximate Analysis.

38. Fixed carbon	per cent.	54.85	54.49
39. Volatile matter	" "	34.10	34.80
40. Moisture	" "	2.85	2.20
41. Ash	" "	8.20	8.51

Ultimate Analysis.

42. Carbon	per cent.	75.83	76.41
43. Hydrogen	"	5.16	5.12
44. Oxygen	"	8.31	7.78
45. Nitrogen	"	1.36	1.37
46. Sulphur	"	1.14	.81
47. Ash	"	8.20	8.51

Analysis of Pan Refuse.

48. Combustible	per cent.	27.73	34.85
49. Ash	"	72.27	65.15

Analysis of Stack Refuse.

50. Combustible	per cent.	72.01	68.19
51. Ash	"	27.99	31 81

Fuel per Hour Steaming Time.

52. Actual coal fired	lbs.	2,368	2,450
53. Equivalent coal burned	"	2,004	2,134
54. Combustible burned	"	1,782	1,906
55. Actual coal fired per sq. ft. grate	"	76.3	78.8
56. Equivalent coal burned per sq. ft. grate		64.4	68.7
57. Combustible burned per sq. ft. grate		57.34	61.3
58. Combustible burned pr. sq. ft. heating surface		.947	1.013

LEFT CYLINDER

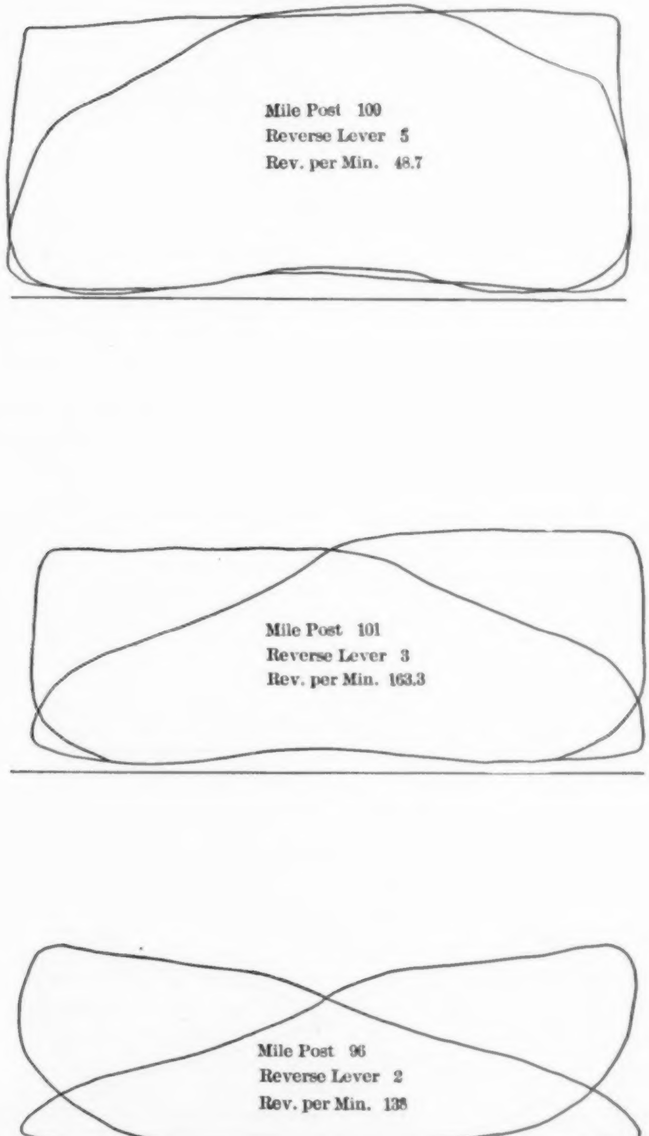


FIG. 191.

Brooks Locomotive No. 257. Run No. 5. Scale of Spring 100.

RIGHT CYLINDER

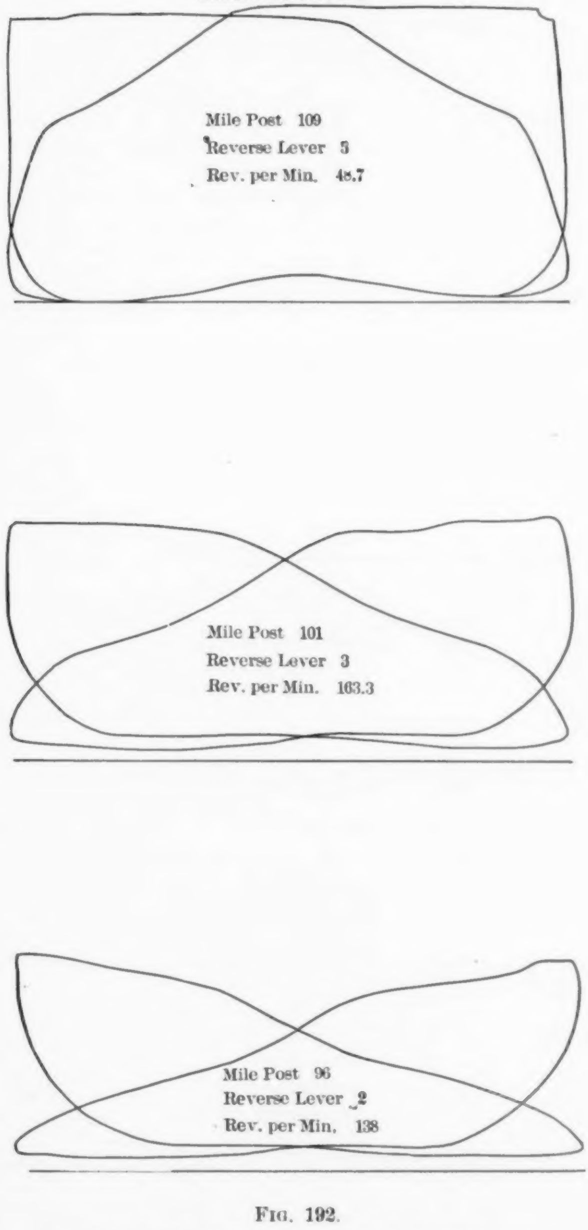


FIG. 192.

Brooks Locomotive No. 257. Run No. 5. Scale of Spring 100.

Calorific Value of Fuel.

H. V. Railway Locomotives.	No. 257.	No. 257.
59. Calorific value per lb. of actual coal by Mahler Calorimeter	13,340	13,394
60. Calorific value per lb. of dry coal	13,731	13,695
61. Calorific value per lb. of combustible	15,006	15,001
62. Calorific value per lb. of stack refuse	10,037	9,505
Quality of Steam.		
63. Quality of steam at dome (dry steam = unity).	.9933	.9960
64. Quality of steam at steam chest	.9771	.9614
65. Quality correction (steam at dome)	.9950	.9979
Water.		
66. *Actual weight of water fed to boiler ... lbs.	73,145	73,481
67. Equivalent weight of water actually evaporated into dry steam	72,780	73,327
68. Factor of evaporation	1.215	1.214
69. Equivalent water evaporated into dry steam from and at 212 degrees	88,420	89,019
Water per Hour Steaming Time.		
70. Equivalent evaporation per hr. from and at 212 deg.	19,505	22,255
71. Equivalent evaporation per hour from and at 212 degrees per square foot heating surface	10.37	11.84
72. Horse-power developed by boiler (34.5 lb. rating)	565.	652.
Economic Results.		
73. Water apparently evaporated under actual conditions per pound of coal as fired	6.82	7.51
74. Equivalent evaporation from and at 212 degrees per pound of actual coal fired	8.24	9.09
75. Equivalent evaporation from and at 212 degrees per pound of dry coal fired	8.48	9.29
76. Equivalent evaporation from and at 212 degrees per pound of combustible minus pan combustible.	9.38	10.42
77. Equivalent evaporation from and at 212 degrees per pound of equivalent coal burned	9.74	10.44
78. Equivalent evaporation from and at 212 degrees per pound of dry coal burned	10.02	10.67
79. Equivalent evaporation from and at 212 degrees per pound of combustible burned	10.94	11.69
Efficiencies.		
80. Efficiency of boiler from combustible minus pan combustible	60.38	67.00
81. Efficiency of boiler from combustible burned	70.43	75.26
82. Efficiency of boiler and furnace from coal fired	59.65	65.55

* Corrected for height of water in boiler and overflow from injector.

Flue Gas Analysis by Weight.

H. V. Railway Locomotives.	No. 257.	No. 227.
83. CO_2 = Carbon dioxide	18.61	18.33
84. O = Oxygen	5.45	6.13
85. CO = Carbon monoxide	.63	.40
86. N = Nitrogen	75.31	75.14
87. Percentage of air excess	31.8	37.35

Heat Balance.

	B. T. U.	p. c.	B. T. U.	p. c.
88. Loss per pound of coal due to products of combustion	1,618	12.13	1,612	12.05
89. Loss per pound of coal due to air excess	441	3.31	520	3.88
90. Loss per pound of coal due to latent heat	486	3.64	478	3.57
91. Loss per pound of coal due to unburned coal in pan	161	1.21	280	2.09
92. Loss per pound of coal due to unburned coal out stack	1,904	14.27	1,471	10.98
93. Loss per pound of coal due to CO	329	2.47	222	1.66
94. Loss per pound of coal due to radiation, etc.	443	3.32	31	.22
95. Heat used in evaporation	7,958	59.65	8,780	65.55
96. Total heat supplied	13,340	100.00	13,394	100.00

Engine.

Number of cars in train	33	33
Gross load in tons	1,490	1,480
Total miles	74.11	74.45
Total revolutions	27,840	27,900
Average speed in miles per hour	15.08	15.75
Average revolutions per minute	94.	98.3
Steam pressure per gauge	180.5	167.
Steam chest pressure by indicator	118.3	117 7
Initial pressures: Right cylinder—head end	113.7	115.7
" " " " —crank end	114.4	117.4
" " Left cylinder—head end	112.1	114.7
" " " " —crank end	112.8	114.1
Back pressure: Right cylinder—head end	8.03	8.89
" " " " —crank end	7.23	7.96
" " Left cylinder—head end	7.63	8.59
" " " " —crank end	7.83	8.22
Mean effective pressure: Right cylinder—head end	58.7	42.1
" " " " " —crank end	50.8	41.4
" " " Left cylinder—head end	54 4	48.2
" " " " " —crank end	54 4	47 5
Horse-power: Right cylinder—head end	128.1	133.
" " " —crank end	124.3	123.4
" Left cylinder—head end	139.9	152.2
" " " —crank end	134.3	143.
Total	526.6	551.6

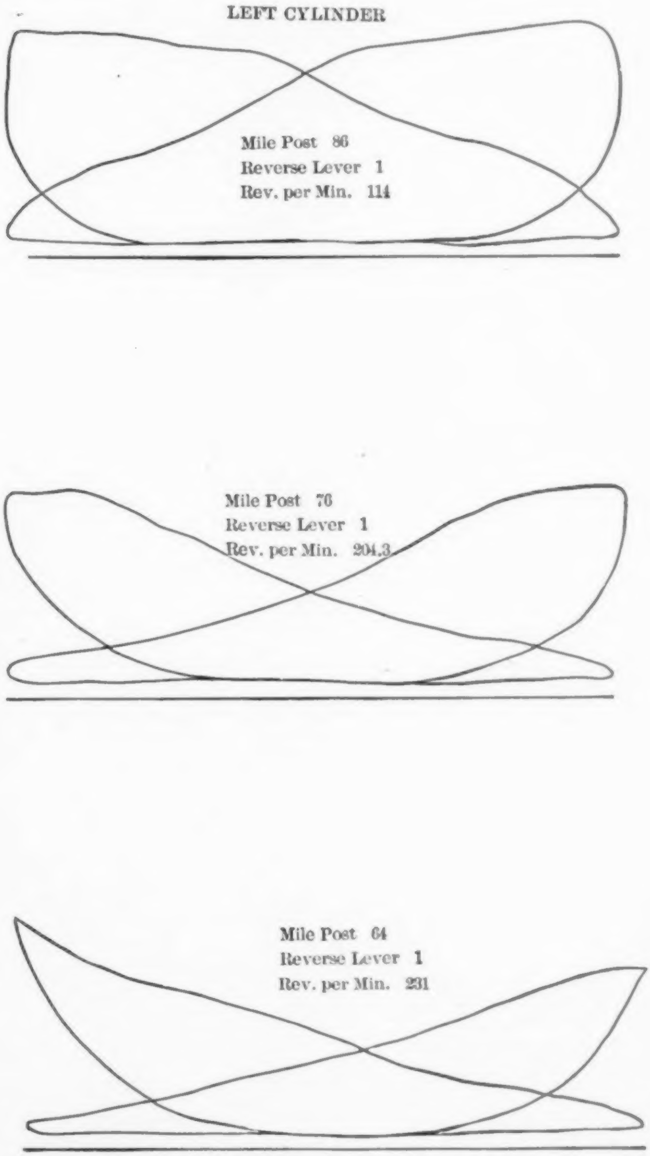


FIG. 193.

Brooks Locomotive No. 257. Run No. 5. Scale of Spring 100.

RIGHT CYLINDER

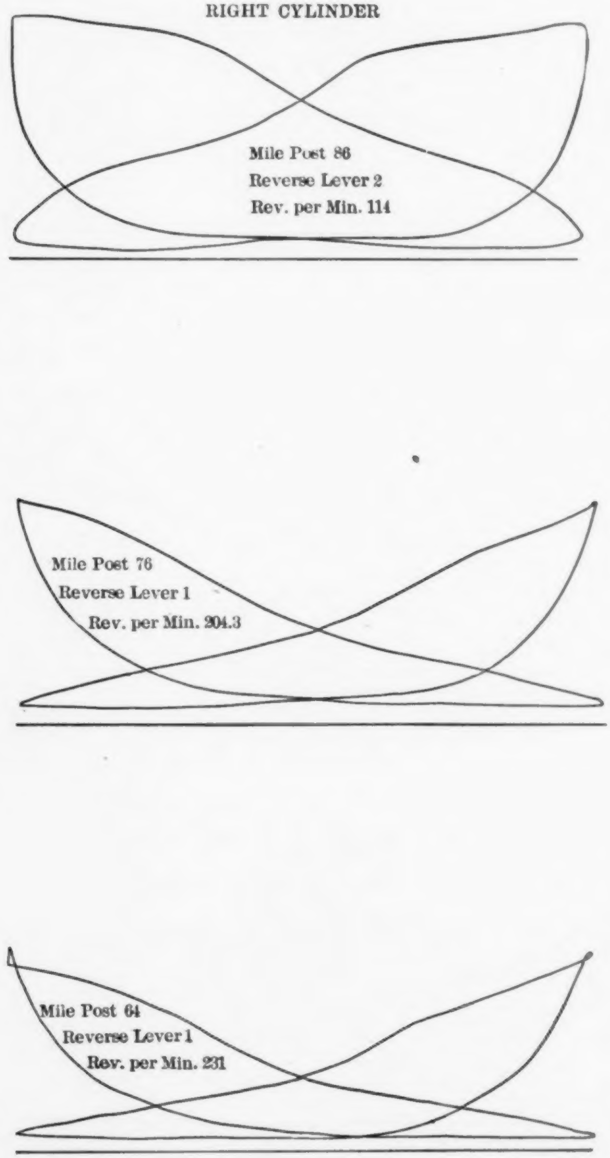


Fig. 194.

Brooks Locomotive No. 257. Run No. 5. Scale of Spring 100.

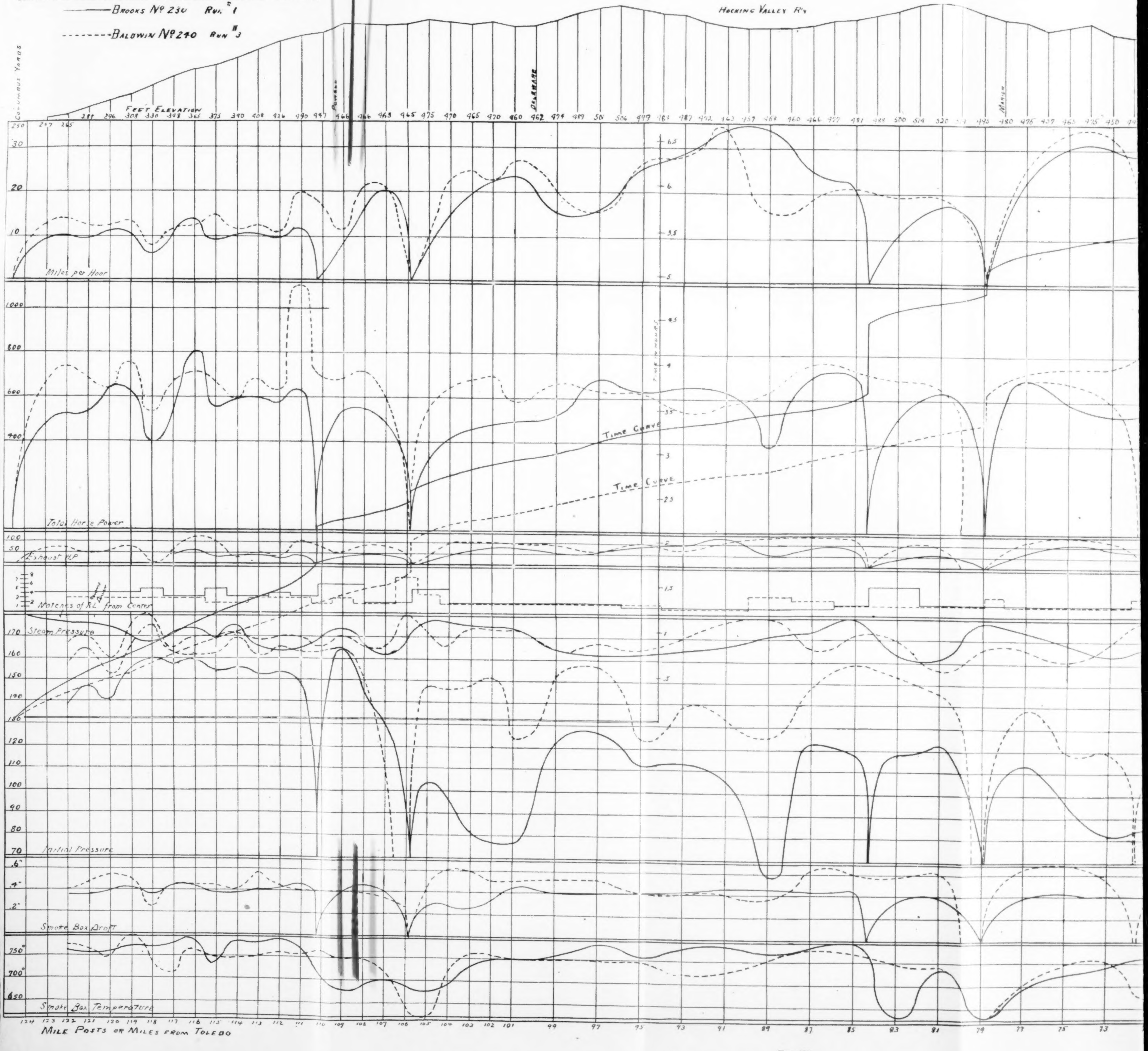

Brooks No 230 Run #1
Baldwin No 240 Run #3
Hocking Valley Ry
Columbus Yards
Powell
Delaware
Marion
Feet Elevation
Miles per Hour
Time Curve
Time Curve
Total Horse Power
Exhaust H.P.
Notches of R.L. from Centre
Steam Pressure
Initial Pressure
Smoke Box Draft
Smoke Box Temperature
Mile Posts or Miles from Toledo

Fig. 195.

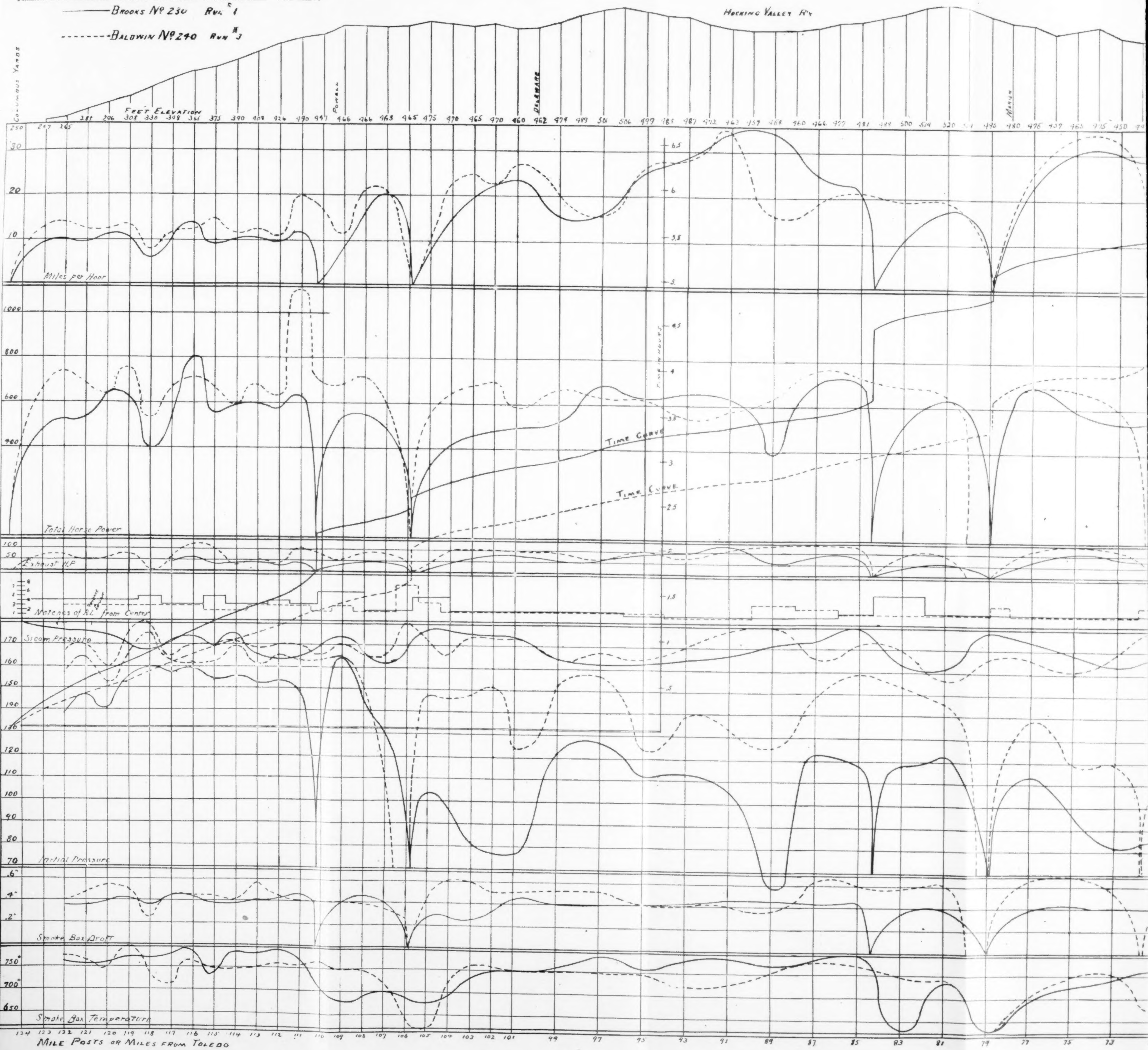
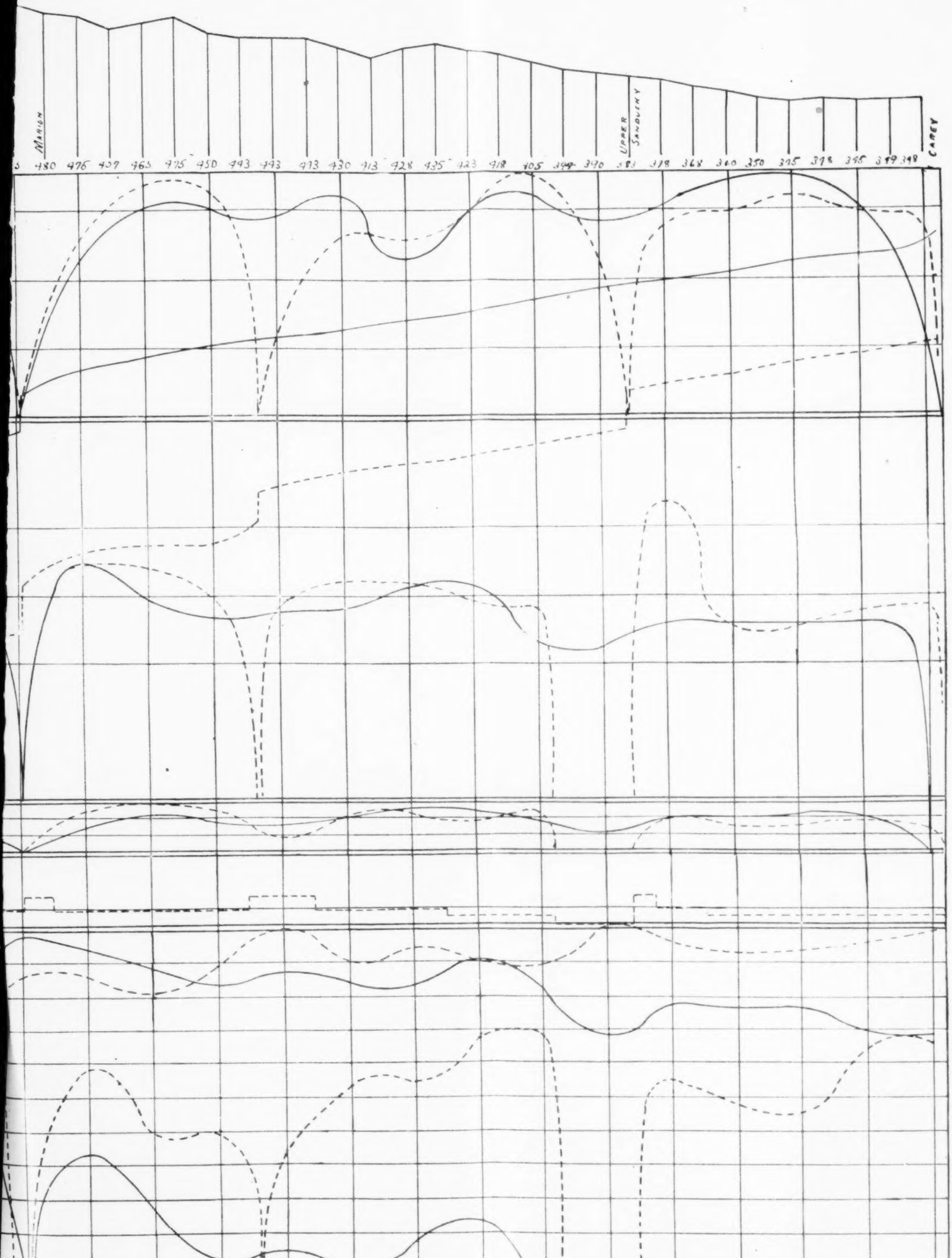

Brooks No 230 Run 1
Baldwin No 240 Run 3
Hocking Valley Ry
Columbus Yards
Feet Elevation
Powell
Delaware
Marion
Miles per Hour
Total Horse Power
Time Curve
Time Curve
Exhaust H.P.
Notches of R.L. from Center
Steam Pressure
Initial Pressure
Smoke Box Draft
Smoke Box Temperature
Mile Posts or Miles from Toledo

Fig. 195.


Marion
Upper Sandusky
Carey
79
77
75
73
71
69
67
65
63
61
59
57
55
53
51

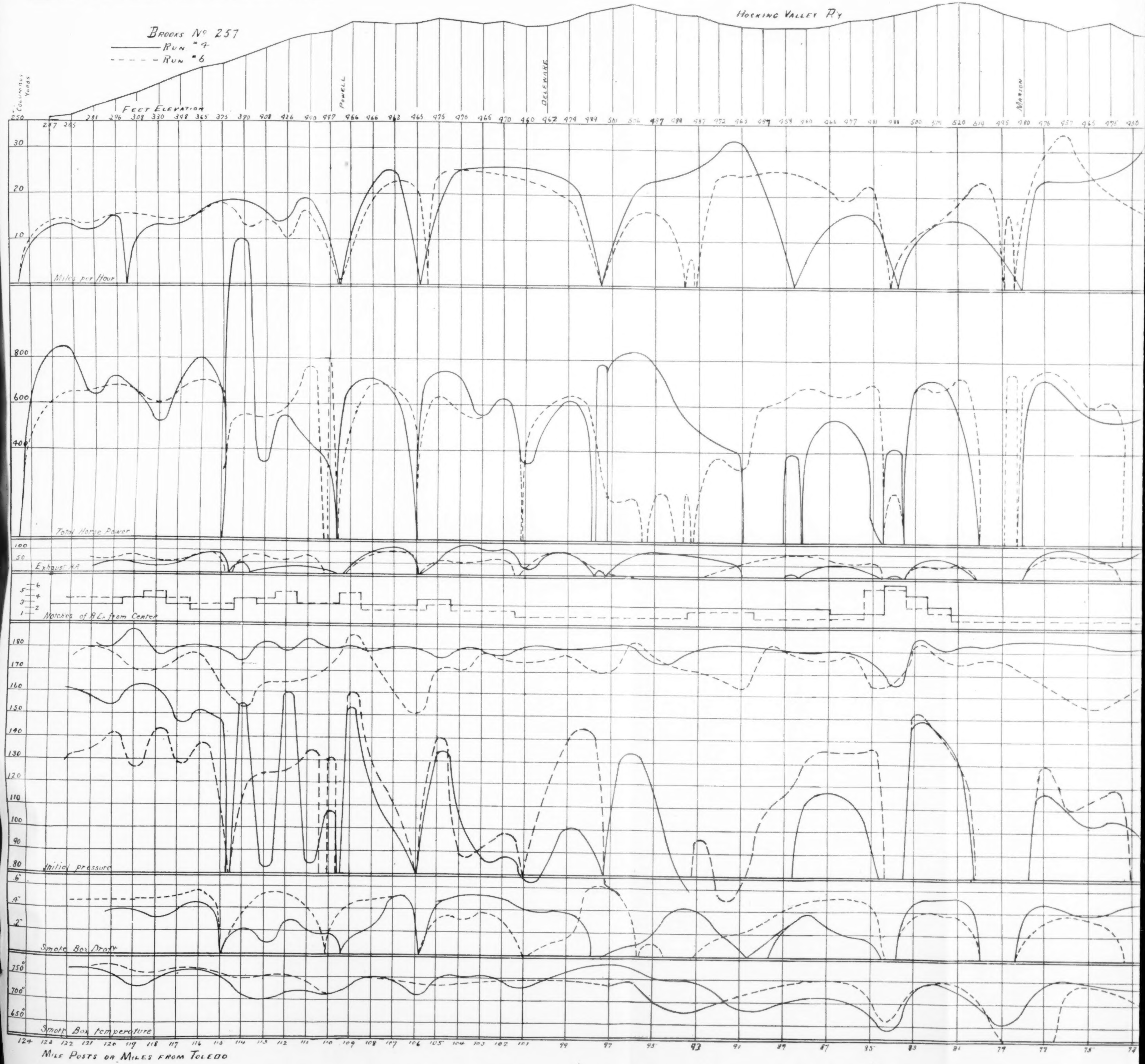
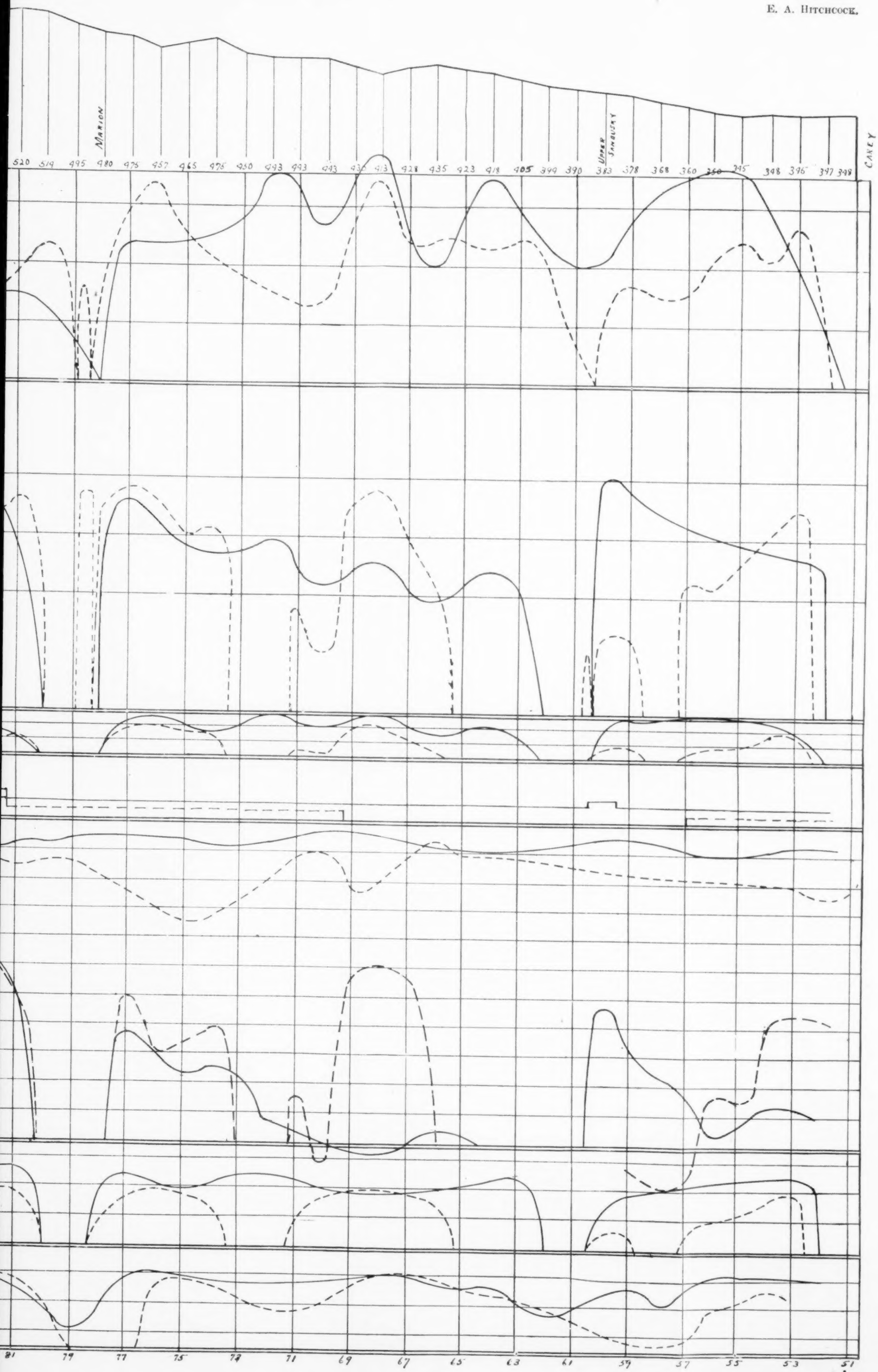

Hocking Valley R'y
Brooks No 257
Run #4
Run #6
Columbus Yards
Powell
Delaware
Marion
Feet Elevation
Miles per Hour
Total Horse Power
Exhaust H.P.
Notches of R.L. from Center
Initial pressure
Smoke Box Draft
Smoke Box temperature
Mile Posts or Miles from Toledo

Fig. 196.


Marion
Upper Sandusky
Carey
520 519 495 480 475 457 465 475 450 443 443 443 436 413 428 435 423 418 405 399 390 383 378 368 360 350 345 348 396 397 398
81 79 77 75 73 71 69 67 65 63 61 59 57 55 53 51

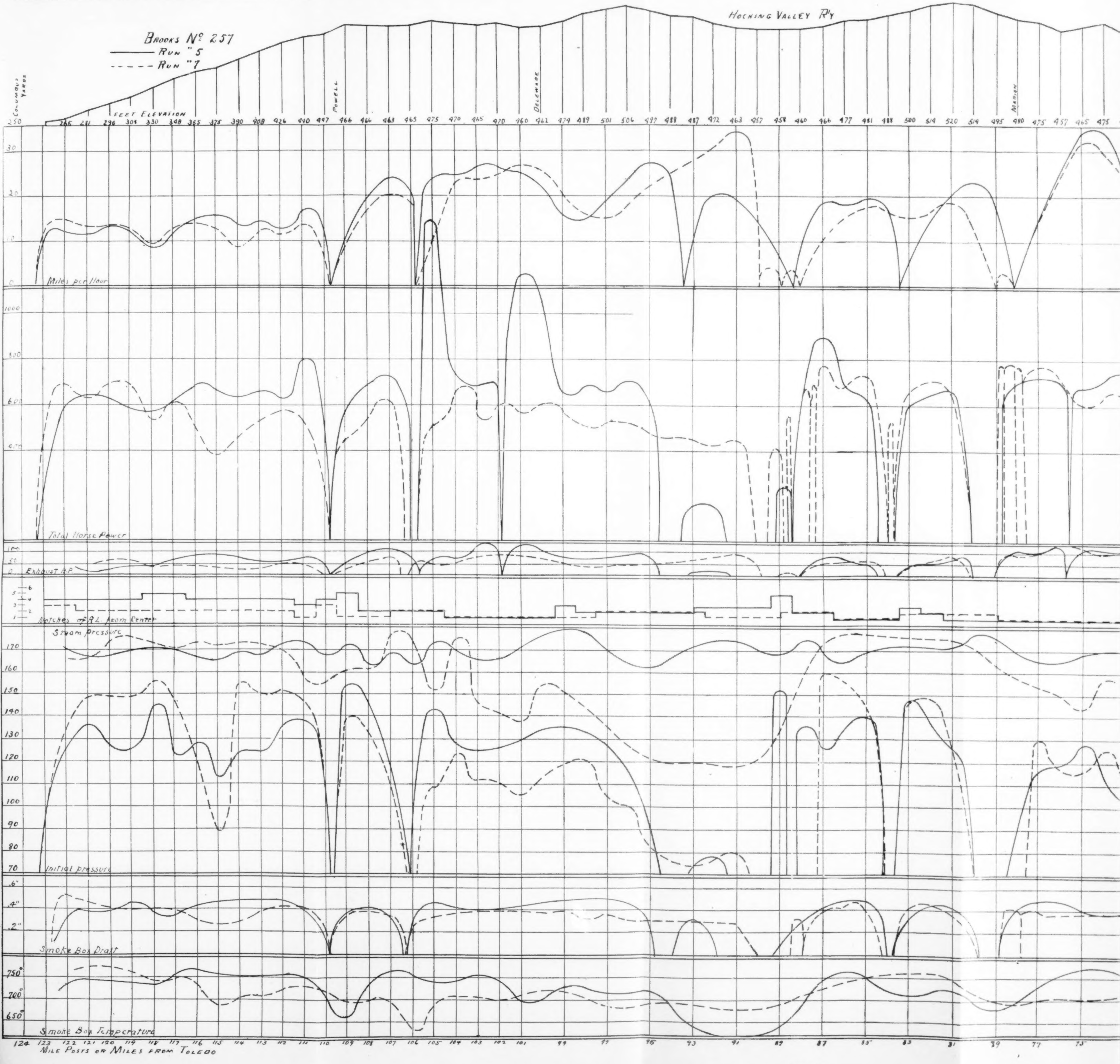
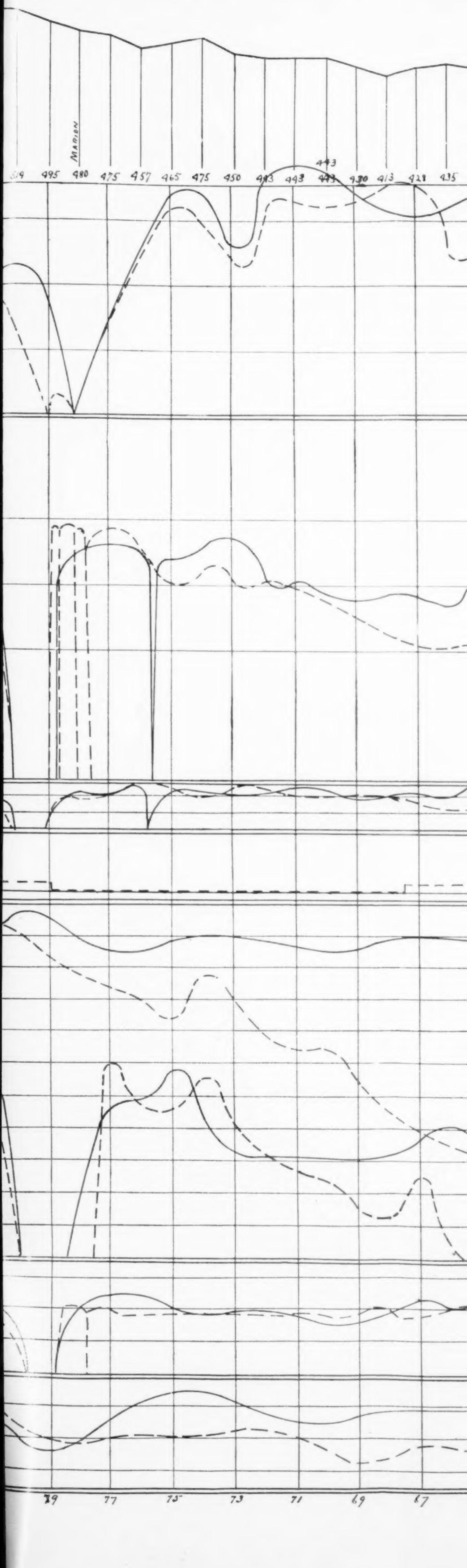
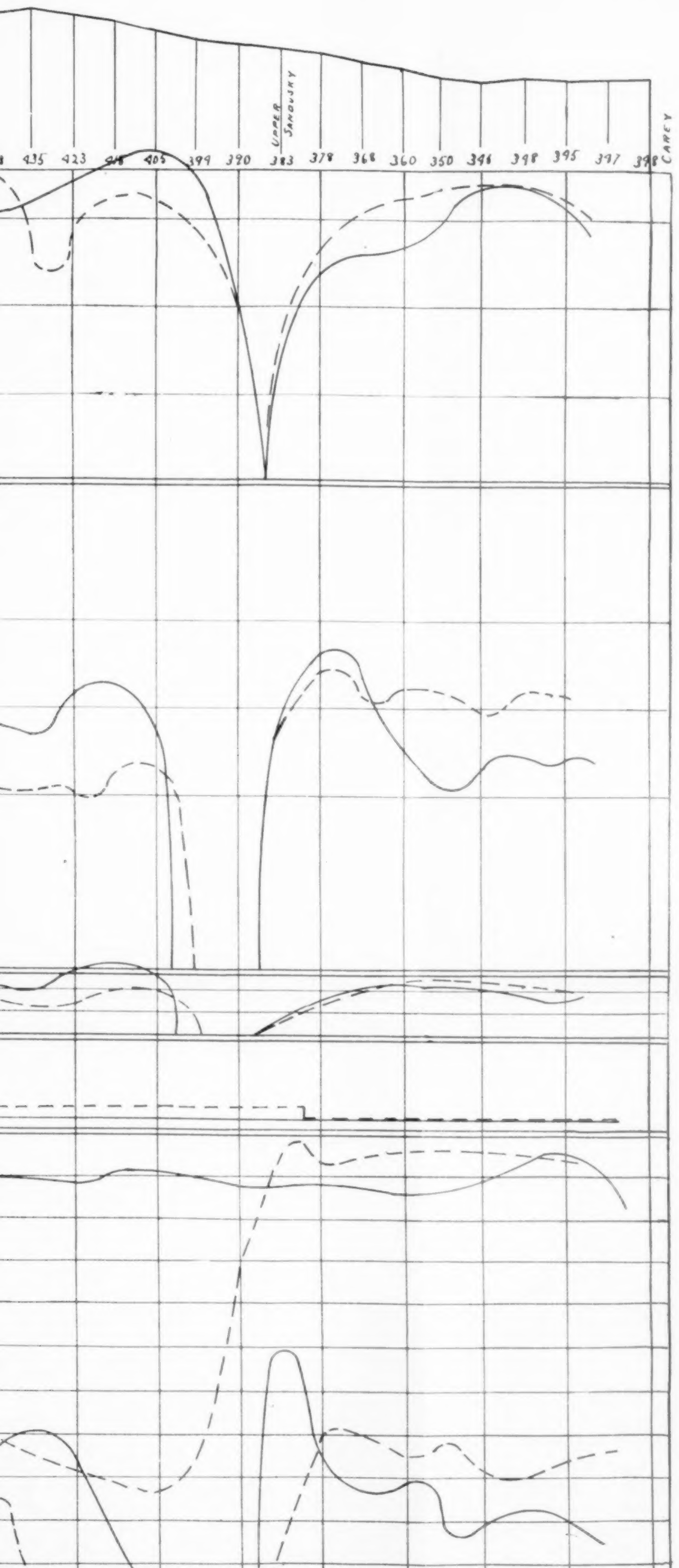

Hocking Valley R'y
Brooks No 257
Run "5
Run "7
Columbus Yards
Powell
Delaware
Marion
Feet Elevation
Miles per Hour
Total Horse Power
Exhaust H.P.
Notches of R.L. from Center
Steam Pressure
Initial pressure
Smoke Box Draft
Smoke Box Temperature
Mile Posts or Miles from Toledo

FIG. 197.



MARION
519
495
480
475
457
465
475
450
443
443
443
443
430
413
428
435
79
77
75
73
71
69
67


UPPER SANDUSKY
CAREY
435 423 405 399 390 383 378 368 360 350 346 398 395 397 398
65 63 61 59 57 55 53 51

H. V. Railway Locomotives.	No. 230.	No. 240.
Exhaust horse-power	77.42	70.2
Safety valve: Number of times popped	59	35
" " Number of seconds on	6,960.	3,654.
" " Lbs. of steam escaped	3,680	1,928
Steam lost at calorimeters: Dome	480	368
" " " " Valve chest	417	263
Dry steam lost by pop and calorimeters	4,550	2,554
Dry steam used by engine and air pump	68,230	70,733
Total hours throttle open	4.53	4.0
I. H. P. hours	2,386.	2,206.4
Lbs. dry steam per I. H. P. per hr. including air pump.	28.6	32.05
Total strokes of air pump	6,000	6,889
Dry steam used by air pump ... lbs.	876.	1,005.
Lbs. dry steam per I. H. P. per engine alone	28.24	31.6
Ton miles	110,423.	108,324.
Lbs. coal per ton mile ... lbs.	.0971	.0905
Lbs. steam per ton mile ... lbs.	.610	.643

30. It will be seen by the general results that there are only four out of the seven trials given. This is due to irregularities creeping in which would effect certain parts of the results, these irregularities being a leaky boiler in two cases and using some of the weighed-up coal for another purpose previous to the trial. However, portions of the results which such irregularities would not effect are given in the curves drawn.

31. These curves show the profile of the Hocking Valley Railroad from Columbus to Carey, nearly due north a distance of 74.5 miles; the speed in miles per hour; total horse-power; horse-power lost, due to back pressure; position of reverse lever; steam pressure; average initial pressure in cylinders; draft in front end and temperature of escaping gases. The areas enclosed by the curves and their respective base line give the averages in the table of general results.

32. During trials No. 1 and No. 3 the per cent. of moisture in the steam as shown by the Barrus calorimeter at the steam chest was slightly less than at the dome, while in trials No. 4 and No. 5 the reverse was the case—dryer steam at the dome than at the chest. Although highly improbable, it was thought that this might be due to the fact that on runs No. 4 and No. 5, the flue gas aspirator received its steam from the dome sampling tube, this tube having a tee looking down to which was connected the pressure gauge and the aspirator, the calorimeter connecting to the horizontal leg of the tee. In trials No. 1 and No. 3 such was not the case, the calorimeter connecting direct to the sampling tube.

In order to satisfy ourselves on this point, a special run was made on the same engine with the same sampling tube and the calorimeter, but with the tee removed and the percentage of moisture shown was practically identical with that obtained on the trials mentioned. In all cases the connections to the calorimeter were well covered, and after the trials the two instruments were tested on the same sampling tube and found to give parallel readings, so that it is evident that the sample of steam at one place or the other was not a representative sample of the steam at that place.

The difference in ash pan draft on trials No. 1 and No. 3 is due to the fact that the pans of the Brooks locomotive are more open, having coarse wire screens at the side.

33. In comparing trials No. 1 and No. 3 the difference in steam consumption is probably due in part to the fact that engine 230 was not working "square" as the cards will show, and also that the piston gland packing of the right side and the valve stem packing and relief valve on the left side leaked somewhat. On the other hand in comparing trials No. 4 and No. 5 where the conditions are practically constant as to engine, crew, load, etc., quite a variation in the steam consumption will be seen, undoubtedly due to the different running conditions as shown by the curves, length of times that throttle was open and time in motion.

34. As regards the boiler performance, comparing the several heat balances, the results obtained show that the "radiation, etc.," loss is a small amount, the average for the four trials being 1.57 per cent. This is to be expected with an internal fired boiler and high rate of combustion. Although this percentage loss would increase as rate of combustion decreases, yet on the locomotive boiler with rate of combustion approaching that obtained in stationary practice, it would still be somewhat less than the same loss as determined in the same way on a boiler with brick setting like the boiler at Ohio State University. (See paper No. 997, vol. XXIV., *Transactions* A.S.M.E.)

35. The greatest loss as shown by the several heat balances is that due to combustible matter passing out the stack. This is dependent upon many conditions, that is, kind and condition of coal, intensity of draft, single or double exhaust nozzle, number of starts and stops, etc. Trials 1 and 3 giving respectively 16.73 per cent. and 12.95 per cent. are with single and double exhaust nozzles and a free burning, non-coking coal, while trials 4 and 5

with same locomotive and having single exhaust nozzle and a coking coal, gave respectively 14.27 per cent. and 10.98 per cent.

36. As stated above the results of several of the trials are not given on account of certain irregularities creeping in which would effect the results as a whole but not in part. For example on trial 7, the boiler leaked somewhat around the fire-box stay bolts when the engine was not in motion, but when under way with hot fire, the leakage practically disappeared. Therefore, the boiler results for this trial with the exception of those which involve the amount of feed water should be practically correct and as trial 7 was on a Brooks engine with Hocking lump coal or a duplicate of trial 1 of the previous year, some of the boiler results given below can be used in comparison with those of trial 1.

Number of trial		7
Running time	hrs.	4.76
Force of draft—front end	ins.	3.77
Temperature of escaping gases	deg. Fahr.	732
Weight of coal fired	lbs.	12,927
Weight of refuse in pan	"	598
Percentage of refuse in pan to coal	per cent.	4.61
Total refuse passing flues	lbs.	2,585
Percentage of ash lost to total ash	per cent.	65.5
Percentage of refuse through stack to coal	"	19.97

Proximate Analysis of Coal.

Fixed carbon	per cent.	48.92
Volatile matter	"	35.11
Moisture	"	6.10
Ash	"	9.85

Ultimate Analysis of Coal.

Carbon	per cent.	66.84
Hydrogen	"	5.17
Oxygen	"	15.10
Nitrogen	"	1.42
Sulphur	"	1.63
Ash	"	9.85

Analysis of Pan Refuse.

Combustible	per cent.	26.55
Ash	"	73.45

Analysis of Stack Refuse.

Combustible	per cent.	67.71
Ash	"	32.29
Calorific value of actual coal by Mahler calorimeter	B. T. U.	11,744
Calorific value of stack refuse by " "	"	10,431

Flue Gas Analysis by Weight.

CO_2 = Carbon dioxide	per cent.	16 56
O = Oxygen	"	6.74
CO = Carbon monoxide	"	.84
N = Nitrogen	"	75.86
Percentage of air excess	"	42.1

Partial Heat Balance.

	B. T. U.	Per cent.
Loss per lb. of coal due to products of comb	1,305	11.10
Loss per lb. of coal due to air excess	459	3.91
Loss per lb. of coal due to latent heat	486	4.14
Loss per lb. of coal due to unburned coal in pan	156	1.33
Loss per lb. of coal due to unburned coal out stack	2,082	17.75
Loss per lb. of coal due to CO	421	3.59

The above partial heat balance gives 17.75 per cent. as the loss out the stack which compares very closely with that of trial No. 1, or 16.73 per cent.

DISCUSSION.

Mr. Wm. Forsyth.—This paper relates largely to the performance of locomotive boilers, and its principal value is in the continuous analysis of the waste gases and the exact measurements of coal consumption and of refuse, which have resulted in the heat balance. I believe this is the first time that the heat balance has been applied to the road tests of locomotives, and the author states that the object of the experiments was to obtain data for making a heat balance for locomotives and to make comparisons of the locomotive boiler under actual working conditions with the stationary water tube boiler. Although no such comparisons are presented in the paper we can make a comparative showing, by using figures which are given in the paper for engine 257, and those given for stationary water tube boilers by the same author in his paper at the Saratoga meeting last year, and such a comparison is made in the following table:

Losses Per Pound of Coal.	Stationary Boiler. Water Tube		Locomotive 257. Hocking Valley	
	B. T. U.	*Per Cent.*	*B. T. U.*	*Per Cent.*
Due to products of combustion	962	6.4	1,618	12.13
" air excess	316	2.1	441	3.31
" latent heat	406	2.7	486	3.64
" unburned coal *	301	2.0	2,065	15.48
" carbonic oxide	129	0.8	329	2.47
" radiation	1,819	12.1	443	3.32
Heat used in evaporation	11,100	73.8	7,958	59.65
Total heat supplied	15,033	100.00	13,340	100.00

* Unburned coal in ashpan	161 pounds,	1.21 per cent.
Unburned coal out of stack	1,904 pounds,	14.27 per cent.
	2,065 pounds,	15.48 per cent.

It will be seen that the unburned coal for the stationary boiler amounts to two per cent. of the coal fire, while in the locomotive

it is 15½ per cent., and in some of the tests recorded in the paper it was as high as 20 per cent. The loss due to radiation is given as 12 per cent. for the stationary boiler, and 3⅓ per cent. for the locomotive. On page 578 the figures relating to the loss due to radiation for engine 240 must be incorrect, although when used as printed they make the total 100 per cent. If they are correct we must conclude that the loss due to radiation in one locomotive must be 15 times as great as that of another, when the two engines are working under about the same conditions. I prefer to think that the figure given for engine 257 as loss due to radiation, 3⅓ per cent., is about correct, as it corresponds closely with that obtained in the experiments made on the Chicago and North-western road in 1898, on boiler coverings, the results of which are given in the proceedings of the Western Railway Club, Vol. XI. I have also made a comparison of the analysis of the flue gases from the stationary boiler and those from the locomotive 257 (see table), and it is surprising to see that they are almost

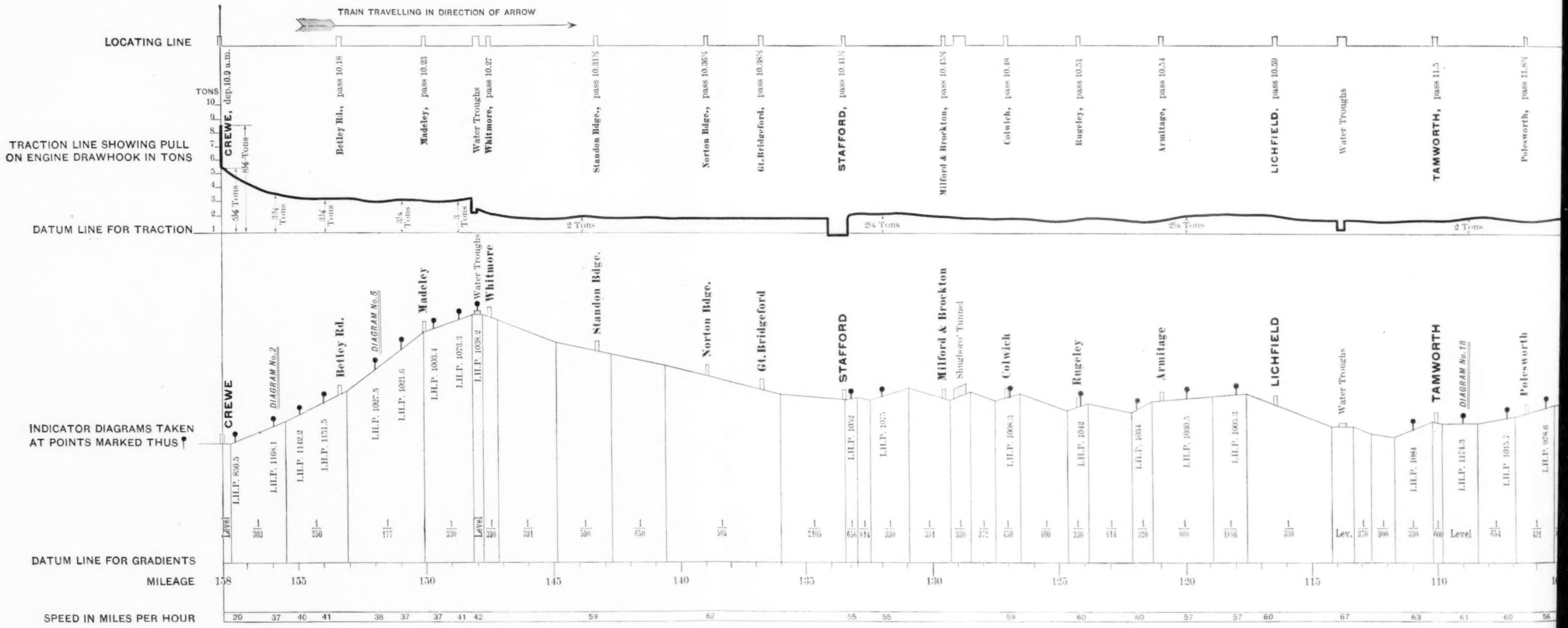
	Water Tube.	Locomotive 230
Carbonic dioxide, CO_2	19.19	18.61
Oxygen, O	5.95	5.45
Carbon monoxide, CO	.16	.63
Nitrogen	74.70	75.31
	100.00	100.00

identical, especially as the percentage of air excess is given as 31.8 for the locomotive, and it must have been very much less for the stationary boiler working under much lower draft in the chimney. It is also surprising to find where the analysis of the flue gases are so nearly identical that the loss due to the products of combustion should be 6.4 per cent. for the stationary boiler, and about double that for the locomotive. It is possible that the author may give a satisfactory explanation of these discrepancies. The performance of these locomotives as here recorded shows that they were not in good working condition, and they are not representative of the best American locomotive practice. From an inspection of the diagrams it is evident that the valve motion was not properly adjusted, and the draft appliances in the front end could not have been of the most approved design. The cards show, as the author states, that the cut-off front and back was not properly equalized. The exhaust line is especially bad and shows a back pressure abnormally high, probably double what it should be for the slow speeds under which the engines

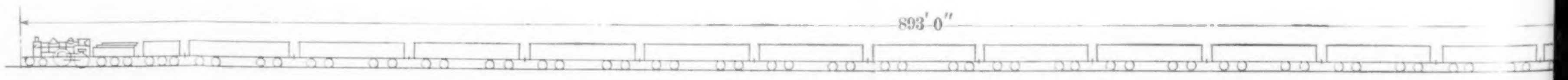
were running. This high back pressure was probably due to two causes: first, to a smaller exhaust nozzle than would have been necessary with a properly designed draft appliance; and, second, to the lack of inside clearance in the valve. It is also evident from the hump at the middle of the exhaust line that an exhaust base similar to that recommended by the Master Mechanics' Association was not used, and this back pressure could have been reduced by using $\frac{1}{16}$ or $\frac{1}{10}$ inside clearance in the valve instead of line and line. I place some importance to the bad effect of this high back pressure, because in the case of engine 240 it amounted to 80 horse-power, which was 12 per cent. of the total power of the engine. It is interesting to see that the ratio of air space to grate area was 47 per cent. for one locomotive and 36 per cent. in the other, a difference of 11 per cent. without any perceptible difference in the economic results. Since the Hocking Valley Railroad has consented to make public in this way, the bad performance of its engines I trust that the same painstaking care which has been used in making the experiments here recorded, has given to that company some valuable suggestion as to how its locomotive performance may be improved.

Mr. A. Bement.—I do not think it proper to consider all water meters as unreliable. One type I have found to be quite accurate, and another type, used extensively in waterworks, is certainly a reliable device.

The heat loss by radiation, etc., as given is extremely small for a locomotive. I would expect it to be greater, and am inclined to believe that there was more moisture in the steam than shown by the sample entering the calorimeter; if this is true it would decrease the efficiency and increase the loss, and as the radiation is obtained by difference as a remainder it would necessarily appear larger. My experience is that it is difficult, if not impossible, to get an average sample of steam into the calorimeter, and with boilers worked at large capacity the tendency is for moisture to appear lower than it is. From a chemical standpoint the term combustible as applied to the fuel minus moisture and ash is not correct, because the constituents are carbon, hydron, sulphur, oxygen and nitrogen, the latter two not being combustibles. In place of combustible it is preferable to designate the fuel minus moisture and ash as pure coal; therefore, the amount of evaporation per pound of combustible would necessarily be greater than per pound of pure coal.

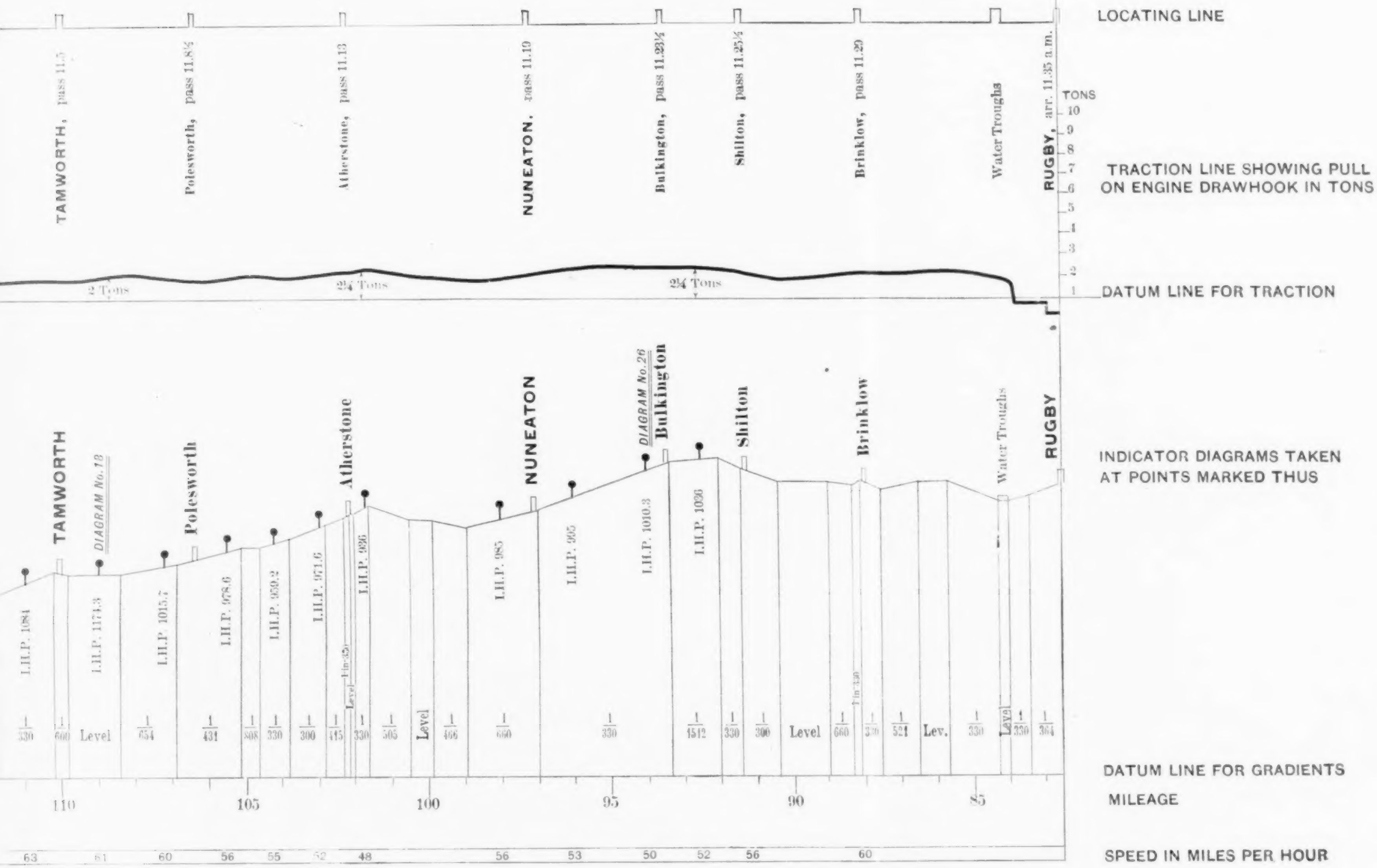


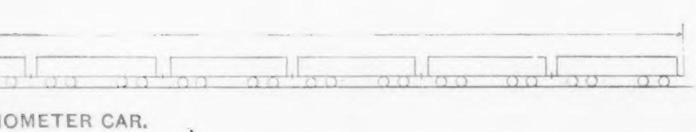
COMPOSITION AND WEIGHT OF TRAIN.	T. C. Q.
4, Coupled Passenger Engine "Precursor"	59 — 15 — 0
3000 Gallon Tender,	37 — 0 — 0
Dynamometer Car,	11 — 0 — 0
15—50 ft. Coaches,	363 — 14 — 3
TOTAL WEIGHT =	471 — 9 — 3
Weight of Train exclusive of Engine and Tender. =	374 — 14 — 3



TRAIN STEAM HEATED AND ELECTRICALLY LIGHTED IN ALL VEHICLES EXCEPTING THE DYNAMOMETER CAR.

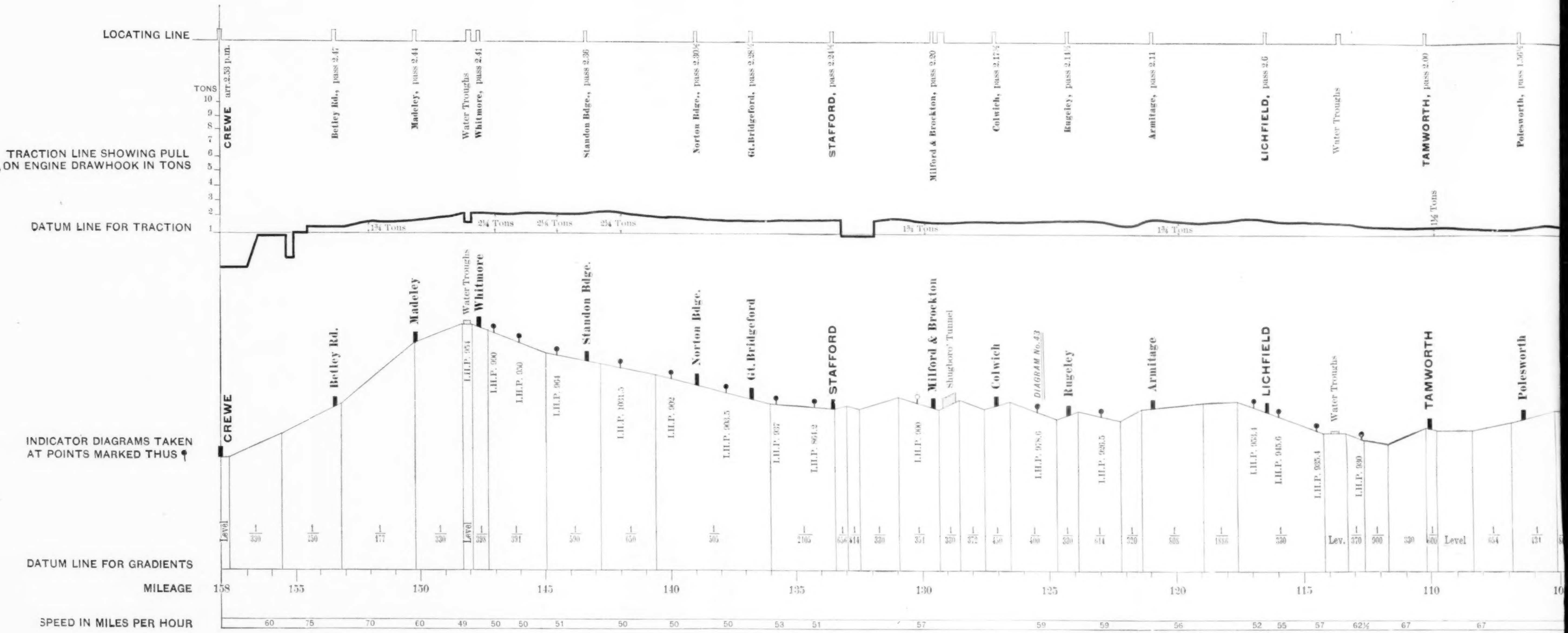
FIG. 198.





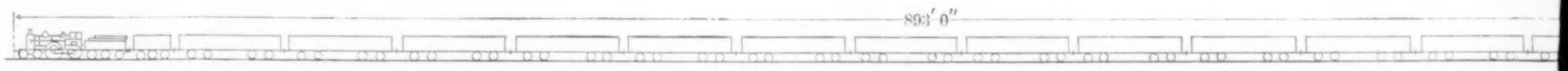
LONDON AND NORTH WESTERN RAILWAY
4-Coupled Passenger Engine "PRECURSOR" No. 513.
DIAGRAM SHOWING SPEEDS, DRAWBAR PULL & HORSE POWER DEVELOPED,
DURING TRIAL TRIP BETWEEN CREWE AND RUGBY ON 27th., MARCH 1904.

Eng'd by American Bank Note Co., N.Y.



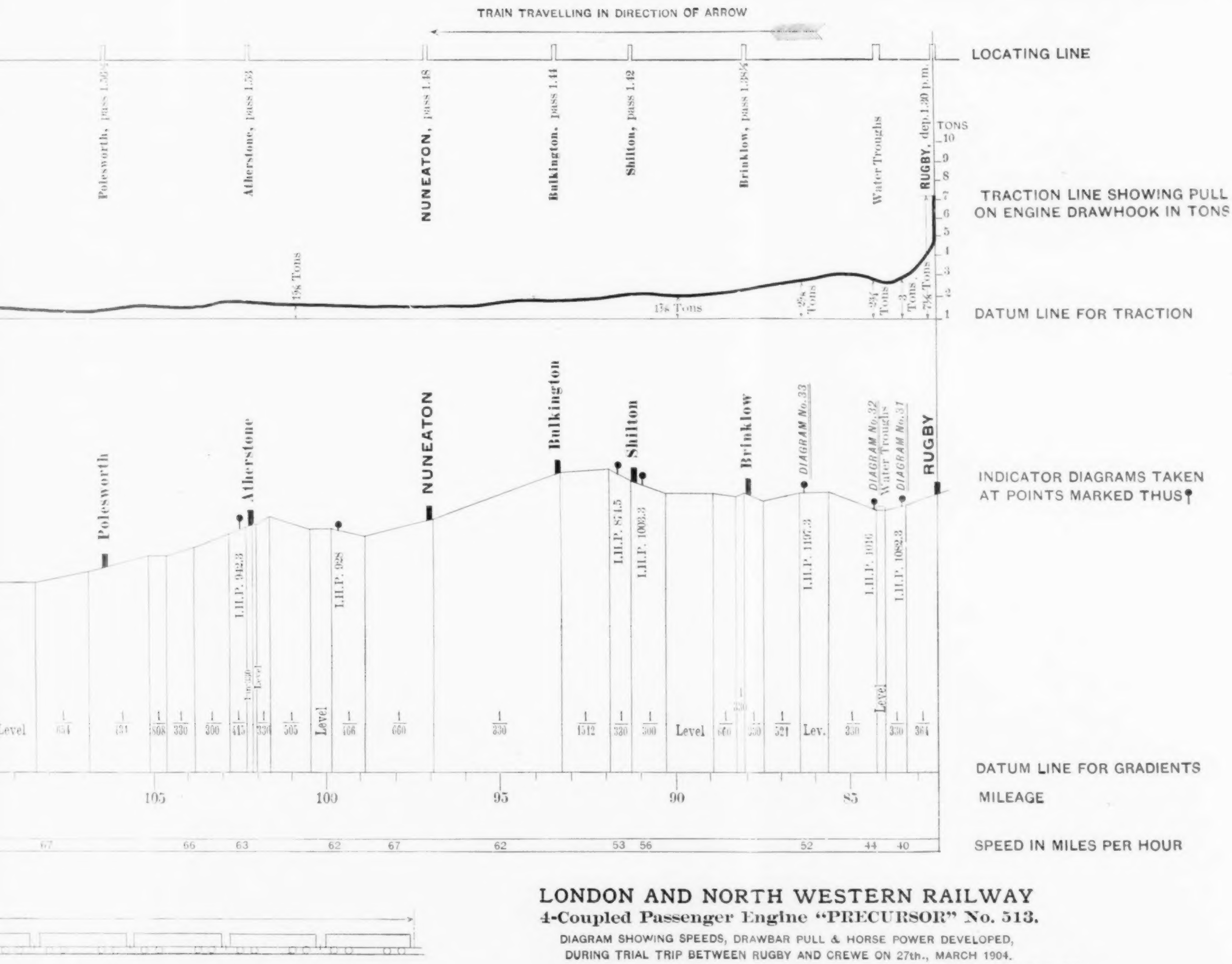
COMPOSITION AND WEIGHT OF TRAIN.

	T.	C.	Q.
4, Coupled Passenger Engine "Precursor"	59	15	0
3000 Gallon Tender,	37	0	0
Dynamometer Car	11	0	0
15–50 ft. Coaches,	363	14	3
TOTAL WEIGHT =	471	9	3
Weight of Train exclusive of Engine and Tender. =	374	14	3



TRAIN STEAM HEATED AND ELECTRICALLY LIGHTED IN ALL VEHICLES EXCEPTING THE DYNAMOMETER CAR.

FIG. 199.



AR.

LONDON AND NORTH WESTERN RAILWAY

4-Coupled Passenger Engine "PRECURSOR" No. 513.

DIAGRAM SHOWING SPEEDS, DRAWBAR PULL & HORSE POWER DEVELOPED,
DURING TRIAL TRIP BETWEEN RUGBY AND CREWE ON 27th., MARCH 1904.

Eng'd by American Bank Note Co., N.Y.

LONDON AND NORTH WESTERN RAILWAY

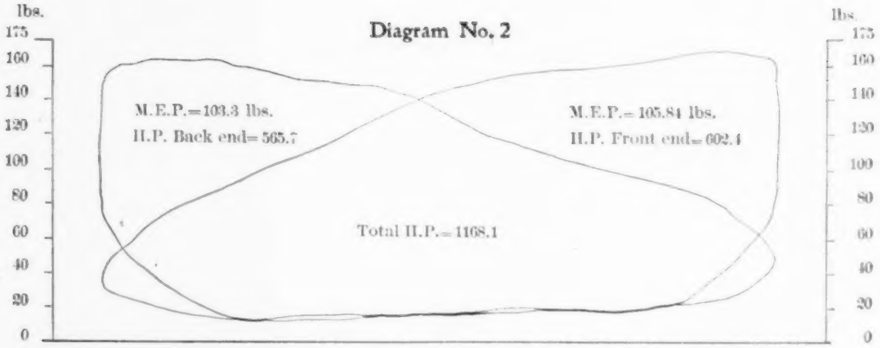
6'-6" Four Wheels Coupled Express Passenger Locomotive "Precursor

(JOY'S VALVE GEAR)

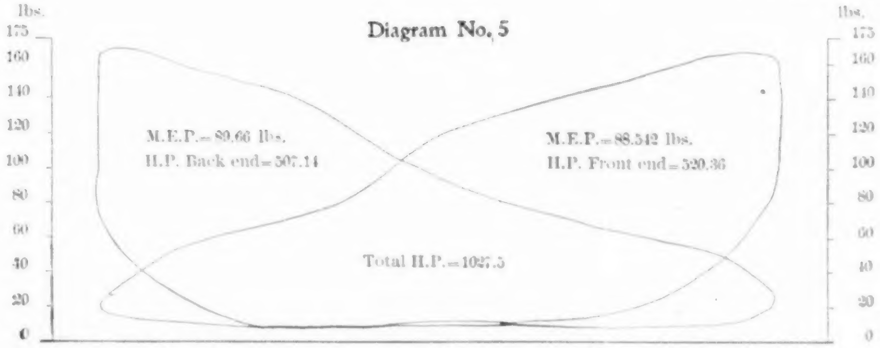
Indicator diagrams taken between Crewe and Rugby whilst working an Experimental tr

Total Weight of train including engine and tender 471 Tons 9 Cwts. 3 Qrs.

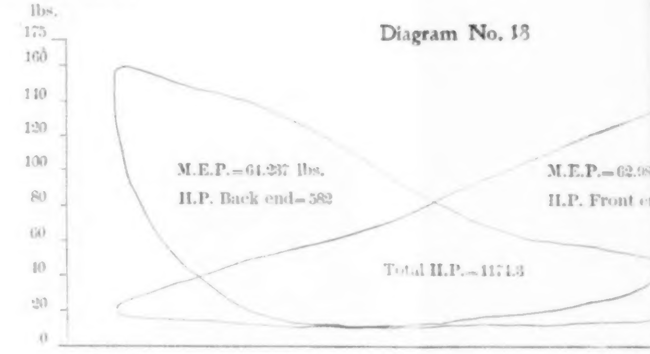
UP TRIP. CREWE TO RUGBY.



Boiler pressure=178 lbs.
Reversing wheel 2 turns back
Speed=37 Miles per hour=153 revs. per min.
Gradient=1 in 330 up
Mile post 156 from London

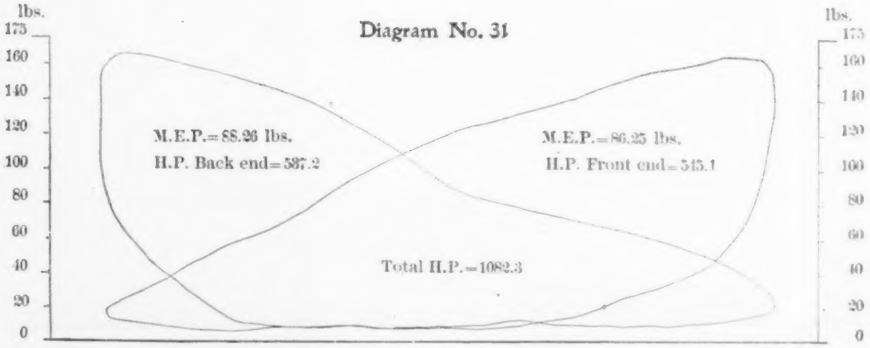


Boiler pressure=174 lbs.
Reversing wheel 3 turns back
Speed=38 Miles per hour=158 revs. per min.
Gradient=1 in 177 up
Mile post 152 from London

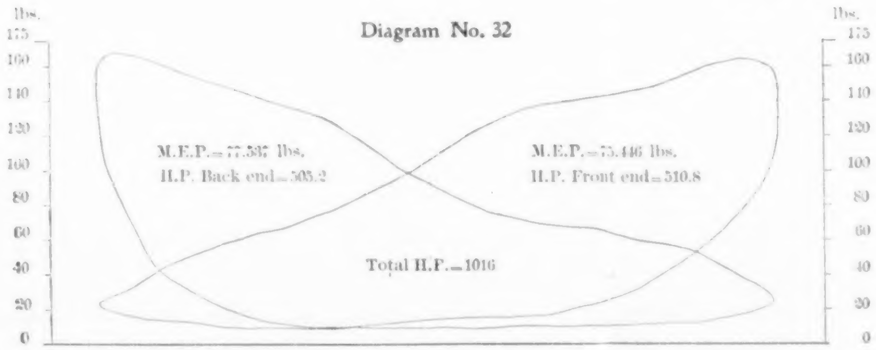


Boiler pressure=175 lbs.
Reversing wheel 3⅛ turns back
Speed=61 Miles per hour=253 revs. per min.
Gradient Level
Mile post 109 from London

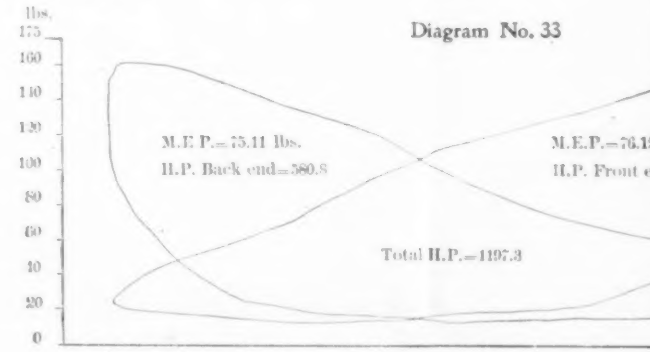
DOWN TRIP. RUGBY TO CREWE.



Boiler pressure=175 lbs.
Reversing wheel 3 turns back
Speed=41 Miles per hour=170 revs. per min.
Gradient=1 in 330 down
Mile post 83¾ from London



Boiler pressure=175 lbs.
Reversing wheel 3 turns back
Speed=44 Miles per hour=182 revs. per min.
Gradient=1 in 330 up
Mile post 84½ from London



Boiler pressure=174 lbs.
Reversing wheel 2¾ turns back
Speed=52 Miles per hour=216 revs. per min.
Gradient Level
Mile post 86¼ from London

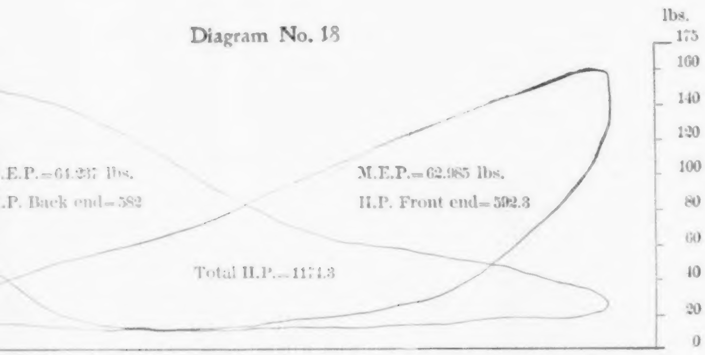
Fig. 200.

N RAILWAY

comotive "Precursor" No. 513

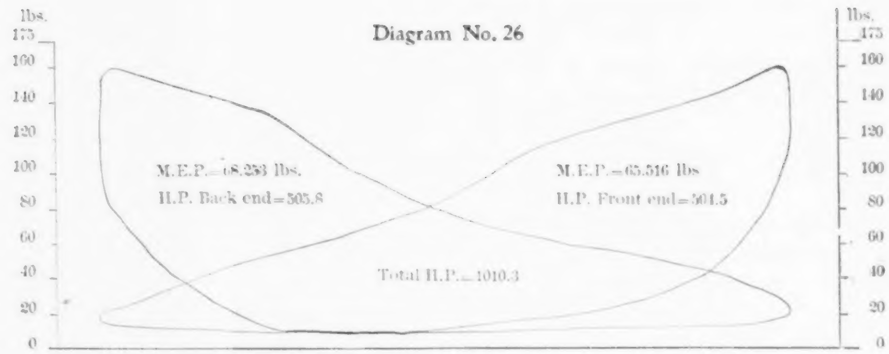
ng an Experimental train on March 27th, 1904

	Tons	Cwts.	Qrs.
d tender	471	9	3

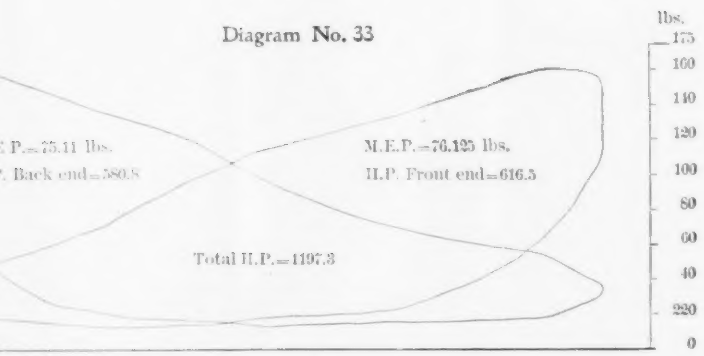


=175 lbs.
l 3½ turns back
s per hour=253 revs. per min.

m London

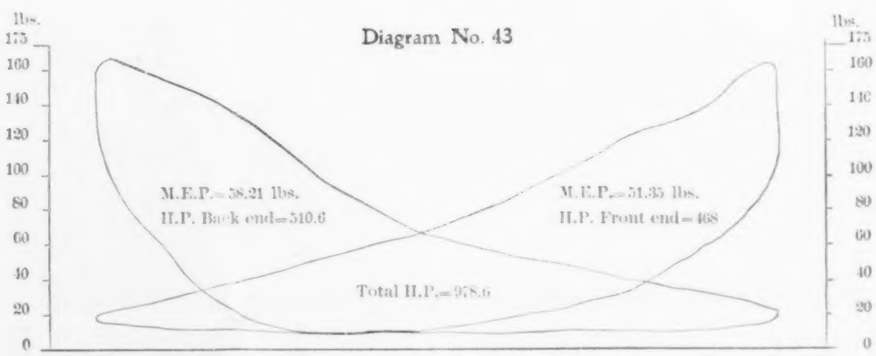


Boiler pressure=173 lbs.
Reversing wheel 2¾ turns back
Speed=50 Miles per hour=207 revs. per min.
Gradient=1 in 330 up
Mile post 94 from London



=174 lbs.
l 2¾ turns back
s per hour=216 revs. per min.

rom London



Boiler pressure=175 lbs.
Reversing wheel 3½ turns back
Speed=59 Miles per hour=245 revs. per min.
Gradient=1 in 400 up
Mile post 125½ from London

Am.Bk.Note Co.,N.Y.

*Prof. E. A. Hitchcock.**—Mr. Forsyth calls attention to the difference in radiation, etc., for engines, as he supposes, 230 and 240. I wish to say that those values were not on different engines, but on the same engine as the statement on page 553 would show, that is, that the trials of 1903 were made on one locomotive only, Brooks No. 257, there being a misprint in inserting the tables. However, the point raised would apply under the actual conditions. That is, the figures indicate that the same locomotive has fifteen times greater loss through radiation, etc., during one trial than during another. To those who have had experience in obtaining boiler heat balances it is not at all surprising to obtain that difference in the unaccountable loss. In fact I should not consider it remarkable in the case of this same locomotive working under practically the same conditions on two trials, to find in computing the heat balances that, in the one case, the radiation loss would come out say 3 per cent., and in the second place come out say zero. Under such conditions, of course, we would not conclude that the radiation loss for this same locomotive in one trial was of an infinite number of times greater than in the second trial.

Since the unaccountable loss in the heat balance is obtained by difference, all the errors due to observation, sampling of coal and refuse, would enter into that factor, and one may expect considerable variation, especially in locomotive work where, on the road, the conditions cannot be controlled as easily as in stationary practice. It is well known that in order to obtain an accurate boiler heat balance the most difficulty is experienced in getting a fair sample of the coal used. This would also apply to the refuse. Now, if there is a considerable variation in the sample of coal, the calorific value as obtained by the calorimeter would vary somewhat from the true value, this variation at once producing quite a percentage of variation in the radiation or unaccountable loss. For example, compare the trials referred to, made on locomotive No. 257, where the radiation loss is fifteen times greater in one case than in the other. If in the second case the true calorific value of the coal used was not 13,394 but was 13,100, a difference of only 2.2 per cent., the radiation loss would then be increased to about 2½ per cent., in which case this loss would have changed from fifteen times to 1.3 times. In fact under certain conditions, in the case of a locomotive, I should not be surprised to see the radiation

* **Author's closure, under the Rules.**

and unaccountable loss come out a minus quantity as long as we determine this factor by difference, containing, therefore, all accumulated errors.

The percentage of air excess is not entirely dependent upon the draft, so that if in a locomotive with a strong draft we obtain, as we did, an air excess of 31.8 per cent., it does not necessarily follow that on a stationary boiler with a lower draft the percentage of air excess would also be lower. This does follow, however, where all conditions as regards boiler setting, the kind and character of coal, and the method of firing remain constant, but in the two cases compared the boilers were different, the coals were entirely different, condition of the coals different, firemen different, and the thickness of fire different.

The loss due to products of combustion alone are computed entirely from the ultimate analysis of the coal, the ash and the temperature of the escaping gases, the analysis of flue gases not entering in any way only in cases where CO is formed. Therefore, it does not follow that, while the flue gases by analysis are nearly identical, the loss due to the products of combustion should also be so. The reasons why in the cases cited the loss from products of combustion for the locomotive boiler is nearly double that for the stationary boiler, is because in one case Hocking coal was used while for the other Pocahontas. This fact, however, would make a comparatively slight difference, the greatest being caused by the difference in flue temperatures, which in the case of the stationary boiler was 499 degrees, while in the case of the locomotive boiler 760.4 degrees. Consider the products of combustion from the locomotive on the same temperature basis as in the stationary boiler, and we would have the air going to the ash-pan at a temperature of 174 degrees, with 499 degrees temperature of gases out of the stack, or a range of 325 degrees instead of 701.40. The locomotive shows a loss of 1,618 British thermal units per pound of coal, or 2.3 British thermal units per degree difference in temperature, which if working through a range of 325 degrees would mean a loss due to products of combustion of 747.5 British thermal units, or 5.6 per cent.

No. 1033.*

TESTING LOCOMOTIVES IN ENGLAND.

BY MR. CHURCHWARD, MR. PETTIGREW AND OTHERS.

I. TESTING PLANT ON THE GREAT WESTERN RAILWAY AT SWINDON.

BY MR. G. J. CHURCHWARD, *Member*,
Locomotive Superintendent of the Great Western Railway.

1. The Great Western Railway Co. have recently put down in their erecting shop at Swindon a plant for testing locomotives. This machine consists of a bed made of cast-iron, bolted on a concrete foundation, with timber baulks interposed for the lessening of vibration. On this bed five pairs of bearings are arranged to slide longitudinally so that they may be adjusted for any centres of wheels that are to be put upon the plant. In these bearings axles are carried having wheels fitted with steel tires on which the locomotive runs. These axles are also fitted with drums on which band-brakes act for absorbing wholly or in part the power developed by the engine. Outside these band-brakes, pulleys having an 18-inch face are provided at each end of the axle for driving link-belts, by which it is intended to transmit the major portion of the power developed by the engine to air-compressors, so that it may not be wasted.

2. The hydraulic brakes will then only absorb just enough power to enable them to govern the speed of the engine. These brakes are actuated by a water-supply from an independent pump, the outlet of this water-supply being throttled either by a stop-valve or by a throttle actuated by a centrifugal governor. This latter device enables the speed of the engine to be set at any required number of revolutions and kept constant.

3. The carrying wheels are 4 feet 1½ inches diameter. The main bearings are 14 inches long by 9 inches diameter. The tire of the carrying wheels is turned to approximately the same section on the tread as the rails in use on the Great Western. This plant is

---

* Presented at the Chicago meeting, May and June, 1904, of the American Society of Mechanical Engineers, and forming part of Volume XXV of the *Transactions*.

intended not only for the purpose of scientific experiment, but also for doing away with the trial trips of new and repaired engines on the main line. It has, therefore, been necessary to make it rapidly adjustable to take engines having wheels of different centres. The main bed is provided with a rack, and each pair of bearings is provided with a cross-shaft having a pinion at either end. These cross-shafts are driven from a longitudinal shaft through suitable clutches. This longitudinal shaft is operated by electric motor and is capable of being reversed. The engine being run over the machine on an elevated frame which carries it on the flanges of its tires clear of the carrying wheels, it is an easy matter to slide these carrying wheels with their bearings till they are vertically underneath the wheels of the engines to be tested. The frame is then lowered electrically and drops the engine into position on the carrying wheels.

4. When running engines on trial trips, it is essential that the bogie and trailing wheels of engines so fitted should be run as well as the driving wheels, in order that the axle-boxes may take a good bearing, and be seen to be in satisfactory condition before handing the engine over for traffic. To accomplish this, the carrying wheels are all coupled together by a suitable arrangement of belts and jockey pulleys. It, therefore, follows that, even when a locomotive having a single pair of driving wheels is run on the plant, all the carrying wheels are rotating and in turn run the bogie and trailing wheels of the locomotive. The jockey pulleys are necessary to retain the proper tension on the belts when the bearings are moved longitudinally.

5. Owing to the varying height of the footplates of different classes of engine, it has been found necessary to provide a firing stage which can be rapidly adjusted vertically. A large coal bunk and weighing machines are provided in connection with this stage. Two water tanks are mounted on the same platform, for measuring the water used when running, these tanks being emptied alternately when a consumption test is being made.

6. Under the platform a dynamometer enables the drawbar pull of the engine to be taken, and this, together with counters on the wheels, will enable the actual drawbar horse-power to be measured, and so compared with coal and water consumption for various classes of engines. As engines of different lengths are to be tested, and of necessity have to be fixed at the trailing end to the dynamometer, it is necessary to have a sliding chimney for

carrying off the steam and smoke from the engine when running. This has been provided in the form of a long box, having a steel plate running on rollers forming its lower surface, which plate carries a large bell-mouthed chimney. This box not only enables the chimney to slide longitudinally, but will also form a receptacle for ashes and any other matter ejected by the engine, which will be retained and can be examined both for quantity and quality.

7. It is hoped that this plant will enable many questions of the relative economy of different classes of engines, either simple or compound, to be settled definitely. The questions of superheating and the efficiency of various forms of smoke-box arrangements might be investigated on it. The effect of various percentages of balancing can be investigated, and, in fact, any of the experiments which are at present being made on the road may be made on this plant, with the great advantage that any engine which may be selected can be placed in position ready for testing, and all connections made in a time probably not exceeding an hour.

II. MEASURED TESTS IN SERVICE.

Instruments and Results Required at a Test.

BY MR. W. F. PETTIGREW, *Member*,
Locomotive Superintendent of the Furness Railway.

8. As a means of minimizing the working expenses of railway working, there is a tendency towards greater weight and length of trains, requiring more powerful locomotives than formerly. It is necessary to use every precaution in designing engines which are most suitable for the traffic to be dealt with, so that the greatest efficiency may be obtained. One of the chief items in the locomotive working expenses is the fuel consumption, which makes it most important to have a complete testing plant, if economical results are to be obtained. Before contracts for coal are accepted, samples should be thoroughly tested as to their relative consumption and cost. In making engine tests the coal should be carefully weighed, and that which is left over should also be weighed and deducted from the total. The following data should be obtained:—

The average Boiler Pressure and I.H.P.
Amount and Temperature of Feed-Water.
Temperature of Gases leaving smoke-box.
Calorific value of the Fuel.

From these the efficiency of the engine and boiler can be found. It is also necessary to know:

The percentage of ash.
Pull on drawbar by dynamometer.
Load in tons.
Number of vehicles.

From these results the relative consumption and, given the prices, the relative cost per consumption can be calculated.

9. *Indicators.*—One of the most useful instruments connected with locomotive testing is the indicator (either the Crosby, McInnes or other); which if properly handled shows a true record of what takes place in the cylinders, enabling any errors to be corrected in the admission, cut-off, release and compression of steam, also the adequacy of ports and steam pipes which may cause undue waste of energy. It is very essential that the valves and pistons should be kept in good order and all losses reduced to a minimum. Various methods for communicating the motion of the piston to the indicator by reducing gears are adopted. The usual method is to have a pillar carrying a lever which is connected at the lower end by an oscillating lever to the crosshead. The pipes are led from each end of the cylinder and connected by means of a three-way cock enabling diagrams to be taken from each end of the cylinders. In making the connections to the cylinders, care should be taken to keep clear of steam ports and drain cocks; also to have the pipes well lagged, so that correct diagrams may be procured.

10. *Tractive Effort.*—To obtain the tractive effort, or the power exerted in hauling any given train, the dynamometer car, although expensive, is the most useful. The car, which is generally placed between engine and train, is provided with the necessary apparatus for obtaining a complete record of the actual pull on the drawbar, the speed and other data. From this information the effective horse-power available for hauling the train is obtained, which, compared with that given by the indicator cards, gives the mechanical efficiency of the engine. Various other methods are in use for obtaining the drawbar pull, either by means of a hydraulic cylinder or springs. In the former the cylinder is bolted to the under side of the tender frame, and the space at the end of the cylinder filled with oil. The pressure is transmitted from the drawbar through the medium of the oil to a gauge, which is located on the engine platform in view of the attendant. In the construction of the hydraulic cylinder, great care should be taken to reduce

the friction to a minimum. The records which may vary from 1 to 10 tons, should be noted at regular intervals, and if proper attention is paid very accurate results may be obtained.

11. *Speed.*—When the dynamometer cars are not fitted with speed mechanism, counters may be used for the purpose of registering the number of revolutions which are read off at intervals, although for fast running it is impossible to get accurate results. The best method, therefore, is to use a speed indicator and recorder, an excellent type being the Boyer Recorder, which is fixed to the platform of the engine coupled by means of a belt to a pulley on the leading or other axle. The instrument consists of a rotary pump forcing oil into a cylinder producing a pressure dependent upon the speed of the engine, each 1-32 of an inch rise corresponding to a speed of one mile per hour. The drum is provided with a roll of paper, graduated to give the speed and distance travelled, so that the speed at any point on the road, the number of stops, or any shunting that takes place can be seen at once. The machine is also connected by means of a small wire to a gauge in the cab, which is graduated to the number of miles per hour, thus enabling the attendant to see the speed clearly. It is necessary to adjust the machine to the mile-posts along the line.

12. *Feed-water.*—To measure the feed-water various methods are in use, either by graduating the tanks or using a suitable water meter. The former is very often done by passing a staff through the top of tank, also by a water-gauge, one serving as a check on the other. The chief difficulties are that the tender must be perfectly level before the readings are taken. The scale is graduated by admitting known quantities into the tank.

13. Water meters of the Siemens, Worthington or other kind give very reliable results if properly connected. The gauge is graduated to register the quantity in cubic feet as the water is passed out. As a protection to the meter, a back-pressure valve is often placed between the meter and the injector to prevent blowing back. To obtain the exact evaporative power of the boiler, complete records of feed-water temperatures and the water wasted by the injector overflow should be taken.

14. *Temperatures, Vacuums, etc.*—It is also necessary to obtain the vacuums in smoke-box at the base of chimney, level with top of blast pipe, middle of nest of tubes, also pressure in firebox and ashpan. The apparatus generally used consists of a U tube mounted on a graduated board, one leg being connected by a pipe to

the point where the vacuum is to be determined and the other left open to the atmosphere. Temperatures are also taken of the smoke-box gases, which register from 400 to 700 degrees Fahr. A mercurial thermometer may be used for temperatures up to 670 degrees Fahr., beyond this a pyrometer or thalpotasimeter is necessary. To determine the quality of the steam, throttling calorimeters are sometimes used.

15. *Coal.*—The coal measurement is a very important factor. The tender should be cleared off, and the coal to be tested carefully weighed before tipping on to the tender; or, better still, the coal should be put in bags each of 1 cwt. capacity. On completion of the trial the coal left is weighed off and deducted from the total. The quantity used for lighting up should be carefully noted. The temperature of water in boiler at the time of lighting up, also at the drawing fires, actual running time, and time standing should be noted. When the fires are drawn, the ashes from the fire-box, ash-pan and smoke-box are each weighed separately to obtain the percentage of ash. Samples are also taken for calorimeter tests for obtaining the calorific value of each fuel. The average boiler pressure taken at regular periods and steaming qualities should be noted, as bad steaming coal with thin close clinker may cause serious delays to traffic, owing to the frequent cleaning of fires. It is sometimes necessary to make complete analysis, owing to the injurious action of some coal on the fire-box plates.

Results Required in the Testing of Locomotives.

Mean boiler pressure, throughout journey.
Total coal used (exclusive and inclusive of lighting up).
Coal burnt per hour (running time and journey time).
Coal burnt per square foot of grate area per hour of running time and of journey time.
Coal burnt per I.H.P. per hour running time and journey time.
Coal burnt per train-mile, engine-mile, ton-mile, and per lb. pull on drawbar per mile, also per hour.
Calorific value of 1 lb. of coal in B.T.U.
Ashes in smoke-box, in ash-pan, in fire-box, total and percentage.
Total water evaporated.
Water evaporated per hour running time and journey time.
Water evaporated per square foot of total heating surface per hour, both running time and journey time.
Water evaporated per I.H.P. per hour running time.
Water evaporated per train-mile and per engine-mile.
Water evaporated per lb. of coal, exclusive and inclusive of lighting up.

Water evaporated per hour (from feed temperature and equivalent from and at 212° F.).
Maximum I.H.P.
Mean I.H.P. calculated from indicator cards from work done.
Curve of H.P. (Mean height.)
Maximum speed.
Mean speed (exclusive and inclusive of stops).
Actual running time and journey time.
Train and engine miles.
Time from lighting up to taking out fire.
Temperature of water in boiler at time of lighting up.
Maximum and mean vacuum at base of chimney.
Maximum and mean vacuum, level with top of blast pipe.
Maximum and mean vacuum at middle of middle row of tubes.
Maximum and mean pressures through fire-hole door.
Maximum and mean pressures through ash-pan.
Maximum and mean temperatures of smoke-box gases.
Efficiencies of engine, boiler, and engine and boiler combined.
Maximum gradient.
Coal stated includes that used whilst standing for (—) hours.
Maximum and mean pull on drawbar.
Maximum and mean load hauled in tons, exclusive of engine, tender, passengers, and luggage.
Maximum and mean number of vehicles hauled.
Maximum and mean number of journals.
Mean load per journal.
Back pressure at maximum I.H.P. and at maximum speed.
Heat (in B.T.U.) carried away by the products of combustion.
Heat expended in evaporating the water.
Heat lost by radiation, imperfect combustion, and evaporative moisture in coal.
Heat converted into work per minute.
Heat taken up by the feed-water per minute.
Relative consumption of coal based on pull of drawbar.
" " " " " I.H.P.
" " " " " ton-mileage.
" " " " " calorimeter tests.
Relative value of coal = Relative consumption multiplied by cost per ton delivered.

The results obtained should all be shown graphically by means of diagrams, which should give the profile of the line run over.

III. OTHER METHODS OF TESTING.

16. Several locomotive superintendents have written to the Institution describing their methods of testing locomotives in actual service.

17. *Mr. John F. M'Intosh*, of the Caledonian Railway, writes:—"I have yours of 27th ult., and beg to inform you that we have no fixed locomotive testing plant, properly so called, unless a 10-mile gradient of 1 in 75 may be classed as such. All our tests have been confined to those taken in actual running. Indicator diagrams are taken from both cylinders simultaneously at intervals of one minute, the times being given by an observer in the cab working the whistle. The Tabor indicator is used. The number of revolutions is taken at the same times from the revolution counter connected to the crosshead. The steam pressure in boiler and valve chest, the opening of the regulator and position of reversing lever are all noted at one-minute intervals. The quantity of water used is measured at all stopping places by means of a gauge rod, the depth being afterwards read off in gallons from a table. The coal is weighed when placed on the tender, the remainder being again removed and weighed after the trip. In addition, and in order to check the speeds, the times of passing all stations are taken, and also all signal checks.

18. "Tests of this kind are made under difficult conditions for the observers, and it is therefore more difficult to obtain accurate results. At the same time, these are the working conditions for the engine, and these conditions are simply unobtainable in a fixed testing plant."

19. *Sir Douglas Fox* states that, for the every-day testing of new locomotives, friction rollers are used by one or two of our largest locomotive builders, but they are not fitted with brakes, and their sole and only object is to enable an examination of the moving parts of the locomotive to be made, the engine itself remaining stationary. The main object in testing a locomotive is to detect mechanical defects, to see that ample clearances are allowed, and generally that the engine is in a good workable condition. As to what horse-power is developed, what the pull on the drawbar is, how much coal and water is used, the amount or volume of air admitted, temperature of smoke-box gases, etc., etc., no observations are taken. These are data that must be obtained by persons who are specially appointed to undertake this class of work, and who have unlimited time and appliances, and, above all, are not having engines built under contract. No doubt a testing plant will give a considerable amount of information, but a locomotive is subject to such varying conditions of wind and weather, condition of rails, uneveness of road, which are all absent in a

nicely warmed and ventilated laboratory, that all results thus obtained are, in a measure, only comparative.

20. *Mr. T. Hurry Riches*, of the Taff Vale Railway, wrote that all the locomotive departments of the larger railways had elaborate shops fitted with machines for testing materials used in the construction of locomotives. The Great Western Railway and the Great Eastern Railway had carried out a good many tests on locomotives as well as on material. He himself had given special attention to the testing of springs and had erected a powerful machine for this purpose.

The following recent references to locomotive testing may be of interest:—

	Date.	Pages.	Remarks.
Inst. Mech. E.	1893.	199.	Experiments on the Draught produced in different parts of a Locomotive when running (Aspinall).
"Engineering."	9/4/97.	434.	Locomotive Tests (Bryan Donkin).
"	5/11/97.	559.	Report of Tests at Purdue University (Sir Douglas Fox).
"	1/1/98.	20.	Locomotive Performances on L. and N.W.R.
Inst. Mech. E.	1898.	605.	Express Locomotives (President's Address).
"Engineering."	7/7/99.	21.	Non-stop run, Euston to Crewe Dynamometer record with special train.
"	25/1/01.	104.	Locomotive Boiler Tests at Purdue.
"	May 1902.	627, 660, 692.	Tests of Boiler at Purdue University.
Inst. Mech. E.	1904.	—	Discussion on Paper by M. Sauvage. Dynamometer and measured trials in France, and on the London and North Western, the Midland, and the North Eastern Railways.

DISCUSSION.

Mr. T. P. Reay.—An apparatus for trying four-wheel coupled tram engines was erected at Airsdale Foundry, Leeds, in the year 1886. (It consisted of a framework on wheels suited to a 4 foot 8½ inch gauge, so that the whole apparatus could be put upon the ordinary lines and readily removed from place to place.) On the top of the framework which was made with a clear space between plates, 6 feet wide, inside bearings were placed which carried two axles with rollers upon them. These rollers had a diameter of

2 feet, and could be set at any gauge on the axles from 2 feet to 5 feet 6 inches, and the axles themselves could be placed at centers varying from 3 feet to 9 feet. Bearings were arranged close to each roller. They were carried on cross girders and could slide across their supports to any necessary position.

In the center of one driving axle was a brake pulley with a strap and a long lever suitable for measuring the brake-power employed. On the sliding bearings bent plates were firmly fixed to rise as guards above the contact surface of the tram-wheel and the roller, in order to prevent a runaway.

For the purpose of gaining further security a big casting was attached to the back with a slot hole in it, so that a connection could readily be made to the draw gear at any height.

No. 1034.*

APPENDIX IV. TO SIXTH REPORT OF THE ALLOYS RESEARCH COMMITTEE.

THE EFFECTS OF STRAIN AND OF ANNEALING

IN ALUMINIUM, ANTIMONY, BISMUTH, CADMIUM, COPPER, LEAD, SILVER, TIN, AND ZINC.†

BY DR. WILLIAM CAMPBELL,

Barnard Fellow of Columbia University, New York,
late of the Royal School of Mines, London.

1. Since the introduction of the microscope in the study of metals and alloys a great deal of work has been done. Iron and steel have received by far the greatest attention, due to their great importance. The crystalline structure of iron and steel has been carefully worked out by Stead,‡ Osmond§ and many others; that of gold by Arnold,‖ Andrews,¶ and Roberts-Austen and Osmond;** whilst the effects of strain have been studied by Ewing and Rosenhain.†† For a bibliography the reader is referred to the "Metallographist," vol. I (1898), page 168.

---

* Presented at the Chicago meeting, May and June, 1904, of the American Society of Mechanical Engineers, and forming part of Volume XXV. of the *Transactions*.

† Being a summary of various Papers handed in by Sir William Roberts-Austen in October, 1901.

‡ "Crystalline Structure of Iron and Steel," Journal Iron and Steel Institute, 1898, Part I.

§ Bulletin de la Société d'Encouragement pour l'Industrie Nationale, 1895 onwards.

‖ "Influence of Small Quantities of Impurities on Gold and Copper," J. O. Arnold and J. Jefferson, "Engineering," 7 February, 1896.

¶ "Microscopic Structure of Gold and Gold Alloys," "Engineering," 30 September, 28 October, and 9 December, 1898.

** "On the Structure of Metals; Its Origin and Causes," Philosophical Transactions, Royal Society, 1896, vol, clxxxvii.

†† "The Crystalline Structure of Metals," Philosophical Transactions, Royal Society, vol. cxciii, page 353.

2. In this Appendix the structures found on the surfaces of small ingots and buttons have been studied in the case of the metals aluminum, antimony, bismuth, cadmium, copper, gold, lead, platinum, silver, tin and zinc.

3. In order to secure a clean surface the metals were carefully skimmed during casting by means of a small piece of charcoal held over the mouth of the crucible. When cold the ingot was examined under the microscope. In most cases a dendritic structure characteristic of the metal was found at one or more places on the surface. By etching with dilute acid the differences in orientation of the granular crystallization of the metal were shown up. Deeper etching in most cases revealed the structure of the grains or primary crystals themselves. It was often found that casting a thin sheet of metal on a flat stone slab helped to show up the dendritic structure. In some cases casting was performed on a flat iron surface to produce more rapid cooling.

4. The effect of slight strain was observed by examining the surface of a small bar or ingot after bending to a slight angle and straightening again.

5. The effects of great strain were found on examining the structures of ingots, etc., after hammering or rolling. In the case of cadmium, lead, tin and zinc, the metal was rolled or hammered out into thin strips and its structure examined after etching.

6. Annealing was performed by heating the strips or sheet of metal in an air bath or on a hot plate at a temperature of 200 degrees Cent. (392 degrees Fahr.) or under. The specimens were etched and examined, and the effects of annealing observed. Where sections were made the specimens were cut, filed, and smoothed down on Nos. 0 and 00 commercial emery cloth, then rubbed on French emery papers 0 to 0000, and finally polished with broadcloth and rouge. A motor-driven revolving table was used, with interchangeable discs to which the various polishing cloths and papers were attached. The method of photographing the various structures has already been explained.*

7. In the case of very soft metals such as lead and tin, and also of the softer alloys, great care has to be taken when polishing a section. If too much pressure be applied the metal tends to spread and form a film over the surface of the section. Again, when a soft metal or alloy is cut with a file, there is a tendency

* Proceedings, 1901, page 1249.

for particles to stick to the file and tear a surface of the section. After the polishing has been completed and the metal etched, long parallel lines of irregular grains or crystals are found as the result of the strain on the surface. On cutting with a saw a similar crystallization is developed, and unless the metal thus modified is removed during grinding and polishing, this crystallization due to strain appears, on etching, along with the original structure of the metal. Of course the new structure is only on the surface, and can generally be obliterated by deep etching, but it shows that, even under slight pressure, metals and alloys will rearrange themselves.

In the following examples the illumination is vertical unless otherwise stated.

Aluminium.

8. If a bar of commercial aluminium be examined depressions will be found on the surface at those points where the metal has solidified last. In these depressions are seen numbers of dendrites of characteristic form, passing gradually into the normal granular structure. On the sides and base of the bar, where it has cooled in contact with the mould, dendrites of a peculiar leaf-like form are seen standing up above the surface, being the first points of solidification.

9. The structure of aluminium can best be examined when the metal has been cast upon a smooth, cold surface. In this way the whole series of structures from grains to dendrites can be found on the base of the metal.

10. Fig. 201, $\times$ 30 diameters, vertical illumination, shows the usual granular structure. The metal was cast upon a polished iron surface, and was $\frac{12}{100}$-inch thick. The view is of the base, which cooled in contact with the iron surface.

11. Fig. 202, $\times$ 30 diameters, shows a similar surface from a piece of metal $\frac{35}{100}$-inch thick. In this case the grain surfaces are covered with very fine markings, which are found to be the outlines of the dendrites or skeletons which build up each grain.

12. Fig. 203, $\times$ 30 diameters, from a piece of metal $\frac{16}{100}$-inch thick, shows a step further in the development of the dendritic structure which is now pronounced. In the centre can be seen a dendrite of cubic form, whilst around it are seen numerous forms elongated in the direction of their growth, for the central dendrite had evidently been a centre of crystallization.

13. Figs. 204 and 205, $\times$ 30 diameters, taken from the base of a

piece of metal $\frac{16}{100}$-inch thick, are in contrast with the above. Here we see a very fine granular structure, because the crystal grain has grown perpendicular to the surface. These are the secondary crystals or grains of Andrews (and Arnold), and possess a distinct orientation. In Fig. 205 several of these secondary grains have grown to a considerable size. They stand out above the surface, having been the first to solidify.

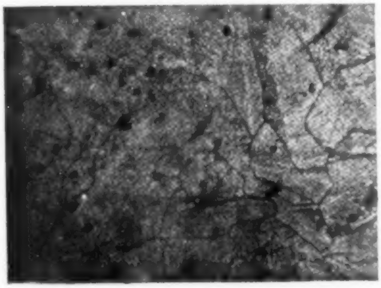
15. In Fig. 206, $\times$ 30 diameters, the maximum development of the dendritic structure is obtained. The specimen was cast on stone, and the view is of the surface. The rate of freezing has been much slower than that of any of the above examples. The dendrites are composed of two axes at right angles, and as a rule only one quadrant of the dendrite has reached perfect development. On etching and examining by oblique illumination, it is seen that the dendrites form the skeletons of the granular structure, as in the previous examples.

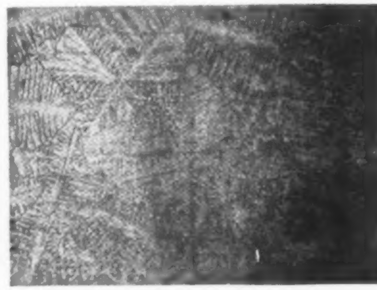
15. Fig. 207, $\times$ 12 diameters, oblique illumination. If a small ingot of the metal be cast and, when partly solid, the remaining liquid be carefully poured off, the internal structure is laid bare. This Fig. shows the surface of such an ingot. Three large grains or crystals are seen, and each is built up of numerous small cubes similarly oriented in the same grain. Under a higher magnification, each cube is found to be built up of very small grains with a distinct orientation. These are the tertiary grains.

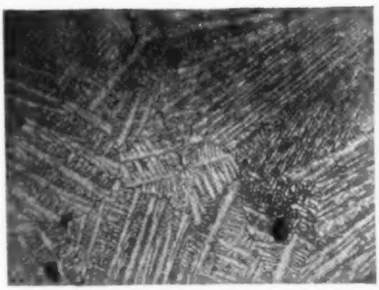
16. When a piece of aluminium, showing the granular structure of Fig. 201, is etched lightly, numerous small pits are eaten out and the surface shows a structure similar to Fig. 202. Under a higher power these pits are seen to have a definite orientation, but their shape is indefinite and obscure, though it seems probable that by etching with a suitable solvent they may be found to be cubic.

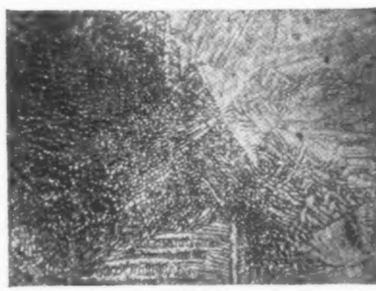
17. When a piece of aluminium is strained, numerous slip lines are formed. So far no broad bands, denoting parallel twinning as the result of strain, have been observed. These slip lines are all parallel to some one or more definite directions, which in many cases were observed to be at an angle of 45 degrees with the axes of the dendrite of the grain; this would coincide with a possible cleavage parallel to the faces of the octahedron. Severe straining tends to show up the granular structure, as was pointed out by Charpy and by Ewing and Rosenhaim.

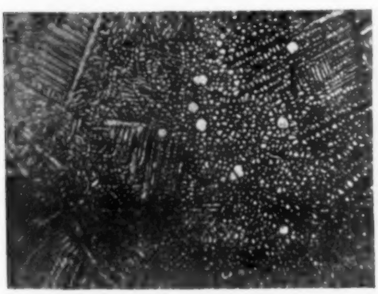
18. Fig. 208, $\times$ 5 diameters, oblique illumination, shows the base of a piece of aluminium cast on iron and severely strained by squeezing.











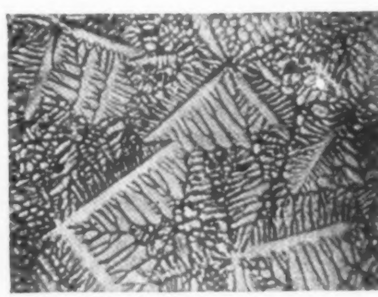


FIG. 201.—Cast × 30 diams.
" 203.— " " "
" 205.— " " "
" 207.—Ingot × 12 "

FIG. 202.—Cast × 30 diams.
" 204.— " " "
" 206.— " " "
" 208.— " and strained × 5 diams.

19. When aluminium is hammered or rolled out, the large grains are broken up and the structure becomes very fine. It is extremely difficult to resolve this structure where rolling has produced a very thin sheet or strip, because here we are apparently dealing with grains which have been elongated. The metal is very difficult to etch. On annealing, the metal loses most of its elasticity and becomes limp. Crystals apparently grow, as in the case of lead, tin, zinc, etc., but as before, the structure is extremely difficult to bring out by etching. Further work is necessary.

Antimony.

20. The beautiful stars met with on the surfaces of large ingots are too well known to need describing. When a small ingot is cast and etched it is seen to possess the granular structure common to most metals. Deep etching reveals the difference in orientation of the primary crystals which are seen to be built up of secondary grains.

21. A piece of large ingot was examined and the surface was found to be covered by comparatively small dendrites composed of two main axes, often at right angles. From each axis grow parallel branches composed of clublike secondary grains, resembling those of bismuth.

22. When fractured, four marked cleavages are found. The most perfect is basal. The three others intersect each other in the basal plane at an angle of approximately 60 degrees, and make with that basal plane an angle of over 140 degrees.

23. When a small cleavage fragment is examined perpendicularly to the basal plane, the effect of the three other cleavages is to produce numerous triangular pits at their intersections. Where only two of the cleavages show, a beautiful rhomboid effect is produced.

24. In many cases, very fine slip bands or parallel twinning was noticed and in a basal fragment this was parallel to the three cleavages.

25. When small ingots were cast, the surface was usually found to be very rough, due to the formation of oxide. On examining the base, however, numerous very small characteristic dendrites were found. On etching, the granular structure was very fine, but each grain was composed of only a very few secondaries. In other words, the primary crystallization is small whilst the secondary is comparatively large. It was noticed that the nearer the

center of the ingot, where the metal was poured in, the smaller was the crystallization. Again, under vertical illumination, by far the greater part of the field is bright and all the bright primaries run into each other. This is apparently connected with the fact that a fracture shows a marked columnar structure perpendicular to the cooling surface.

26. As the metal is so brittle, straining produces fracture almost immediately the ingot is bent. On examining the metal in the region of a fracture only a few traces of slip-bands could be seen on the surface. They were, in most cases, parallel to the line of fracture. In the case of a large commercial ingot, a fine development of parallel slip-bands was found amongst the dendrites on the surface in the region of fracture. Three systems were seen. One was parallel to the direction of one of the axes of the dendrite, the two others making angles of approximately 60 degrees and 120 degrees with it. These slip-bands, as before, were found to be parallel to the cleavages of the antimony.

27. When antimony is broken by a sharp blow, the fractured surface shows a distinctly radial structure, the lines radiating from the point of impact. Under the microscope this structure is seen to follow the true cleavage in minute steps, which are so small that, to the eye alone, the cleavages seem to have been suppressed.

Bismuth.

28. The markedly crystalline structure of bismuth is well shown when a crucible of the metal, with a trace of tin or lead, is allowed to solidify on the surface and the remaining liquid run off through a hole in the side. The under surface of the crust is seen to be composed of many hopper-shaped skeleton cubes. These have a definite orientation and are the secondary crystals.

29. Fig. 209, × 30 diameters. If a small ingot be carefully cast so as to give a clean surface, numerous characteristic dendrites are seen. This Fig. shows such dendrities. Two generations are seen: the large more-or-less parallel bands forming the large granular structure of the ingot, and the finer dendritic structure which in places builds up elongated secondary grains. When a slowly cooled button of the metal is deeply etched the difference in orientation of the large grains is brought out. Each grain or crystal is seen to be built up of secondaries possessing a distinctly cube-like form. They are much coarser than those found in tin, etc.

30. Fig. 210, × 30 diameters, shows the surface of an ingot which has been strained. If an ingot of bismuth be strained by bending, numerous groups of parallel slip-bands or multiple twinnings are set up. The specimen has been lightly etched with hydrochloric acid, which helps to show up the difference in orientation between the parts. When straining is severe, fracture occurs.

31. Fig. 211, × 30 diameters, shows an ingot of bismuth which has been bent until it broke, and it is to be noticed that fracture occurred along straight lines in four directions, but none of these coincides with the two directions in which the banding due to the strain occurs.

Cadmium.

32. The dendrites found on the surfaces of small ingots of cadmium occur in the form of six-rayed stars, of decidedly hexagonal symmetry.

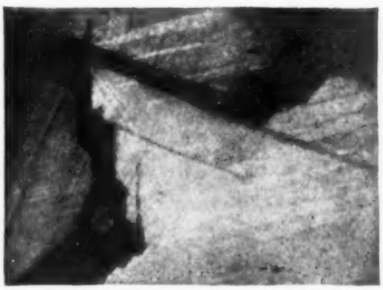
33. Fig. 212, × 30 diameters, shows such dendrites from the surface of an ingot cast in an iron mould. That these dendrites form the skeletons of the coarse granular structure is seen on etching.

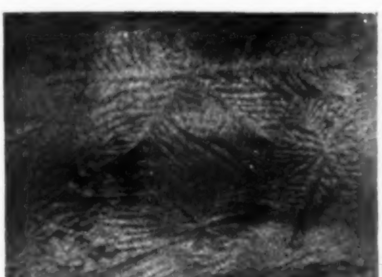
34. Fig. 213, × 30 diameters, oblique illumination, shows the granular structure of cast cadmium, brought out by etching with dilute nitric acid. In one of the larger grains the remains of a dendrite are seen, forming the core of the grain enclosing it. On etching the base of the same ingot, the structure appeared almost identical with Fig. 290, the difference in size of the grains being entirely due to the difference in the rate of solidification. When strained, the metal shows slip-bands as in the case of bismuth, but of a rather finer kind.

35. Fig. 214, × 30 diameters, shows the surface of an ingot which has been strained and then etched with nitric acid. In the central grain, three systems of parallel slip-bands are seen, intersecting each other at an angle of about 60 degrees.

36. Fig. 215, ×30 diameters, shows similar slip-bands set up in the grains by strain. The specimen was etched with nitric acid.

37. When cadmium is hammered or rolled, the coarse crystallization due to the original cooling is broken up and a finer granular structure is produced. The greater the mechanical work upon the metal the finer is the structure produced, down to certain limits, when the grains begin to spread out. A bar of cadmium, $\frac{5}{16}$-inch in diameter, was rolled out into lengths of varying thickness. Pairs of each thickness were cut off, having a length of $2\frac{5}{8}$ inches, and a breadth of 0.45 inch. One strip of each pair was





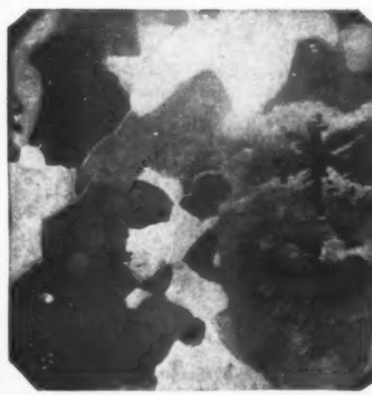


FIG. 209.—Bismuth. Cast × 30 diams.
" 211.—Bi. Ingot, fractured × 30 d.
" 212.—Cadmium cast × 30 d.
" 214.—Cd. cast & strained × 30 d.

FIG. 210.—Bi. cast & strained × 30 d.
" 213.—Cadmium cast " × 30 d.
" 215.—Cd. cast & strained × 20 d.

annealed in an air-bath for seven days at a temperature of 180 degrees Cent. (356 degrees Fahr.) and under. There were five pairs, and they were etched with dilute nitric acid, then dipped in hydrochloric, washed and photographed whilst still wet.

38. In Fig. 216, are seen the five pairs. The thinnest pair are on the left, the thickest on the right, whilst in each pair the unannealed strip is on the left, the annealed on the right. Starting from the left, the thinnest pair, No. 1, have a thickness of 0.018 inch; No. 2 of 0.024 inch; No. 3 of 0.036 inch; No. 4 of 0.049 inch; whilst No. 5, the thickest pair, are 0.075 inch in thickness. The effects of annealing can be seen without the aid of a lens. On examination under the microscope the changes are most marked.

39. Fig. 217, $\times$ 30 diameters, is No. 1 as it came from the rolls, and may be taken as a type of all the rolled strips. The original crystalline structure of the bar has entirely disappeared, and in its place we have the fine-grained structure produced by the breaking down of the original. The effect of annealing was to reduce the flexibility of the strips due to the growth of very large polygonal grains or crystals.

40. Fig. 218, $\times$ 30 diameters, shows No. 5, thickest strip, after annealing. The contract between Figs. 217 and 218 is marked.

Fig. 219, $\times$ 30 diameters, shows No. 4.

Fig. 220 shows No. 3.

Fig. 221 shows No. 1, after annealing. Although the crystals in Fig. 219 are smaller than those in Fig. 220, which is taken from a thinner section, it was noticed that the average size of crystal was greater the thicker the strip.

41. As in the case of tin, zinc, etc., bending the annealed strips beyond the elastic limit sets up slip-bands which in many cases resemble the multiple twinning of plagioclase.

Fig. 222, $\times$ 30 diameters, shows such systems of slip-bands or twinning produced in No. 4 by bending, whilst Fig. 223 shows those in No. 3. In both cases the largest crystals were photographed because they show the effects better. Three systems of parallel slip-bands can be seen in a single crystal.

Copper.

42. The structure of copper and the influence of small quantities of impurities have been studied by Arnold and Jefferson.*

* "Engineering," 1896.

43. Fig. 224, × 30 diameters, obliquely illuminated, shows the surface of a slowly cooled button of copper (99.8 per cent.). Part of two large grains or crystals are seen, built up of smaller or secondary grains with a definite orientation. This is in marked contrast with Fig. 225, × 50 diameters, which shows a surface of electrolytic copper just as it came from the depositing vat. The surface is composed of many polygonal grains or crystals which are primary, and therefore have no regular orientation with respect to one another. Sections of electro-copper have already been shown (Proceedings Institution of Mechanical Engineers, 1901, Plate 206, Figs. 141, 142).

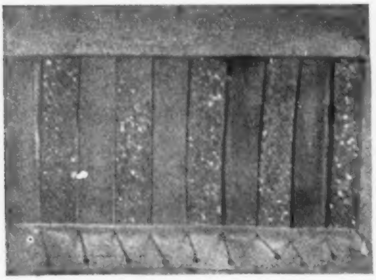
44. Fig. 230, × $1\frac{1}{2}$ diameters, is the top of a button of copper which has been used for calibrating a thermo-couple. The surface shows numerous dendrites composed of two axes at right angles. Etching shows that these dendrites form the skeletons of the grains as before.

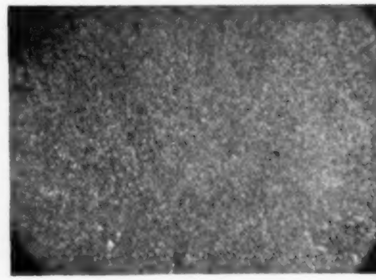
45. Fig. 226, × 30 diameters, shows a cross section of commercial copper (unrefined) cast in the form of a circular bar. The view was taken from near the centre. Large grains are seen, each built up of smaller secondaries, definitely arranged. The centres of each secondary have etched lighter in color than the outside, because the first part to solidify was purer; and as equilibrium was not established mixed crystals have been formed, as explained by Rooseboom (the theory of solid solutions).

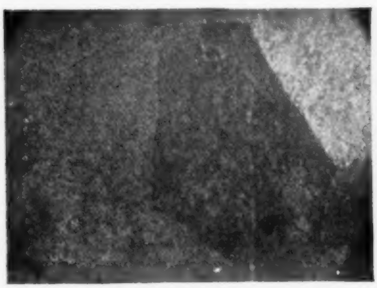
46. When copper is rolled or hammered, the crystals are often found to be twinned. Ewing and Rosenhain have shown that these twins are the result of strain, and persist after annealing at a red heat. Similar twins are found in gold, brass, steel, etc. With great strain the secondary grains elongate in the direction of rolling.

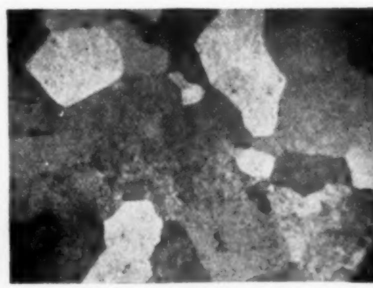
47. Fig. 227, × 30 diameters, shows a vertical section of rolled copper. The secondary grains are elongated in the direction of rolling. With further rolling, these grains become very long and attenuated and may finally break down. Annealing tends to release the strain, makes the metal more pliable, and allows the grains to rearrange themselves.

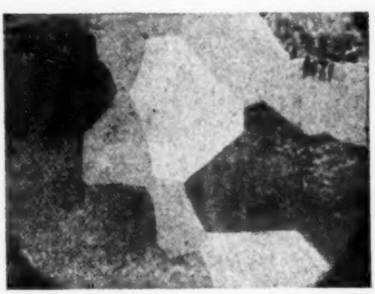
48. Fig. 228, × 50 diameters, shows some copper foil, etched with nitric acid, just as it came from the rolls. The structure is composed of a mass of grains similar to those of Fig. 227, slightly elongated in one direction. All trace of the primary (original) crystallization has entirely disappeared. The foil possessed a

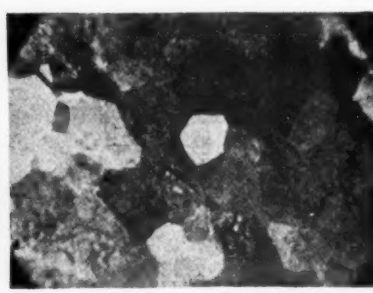








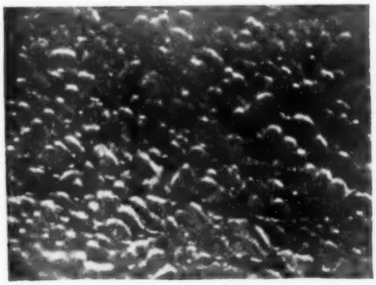


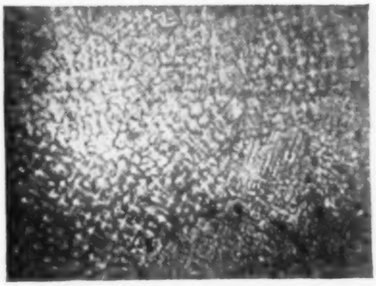


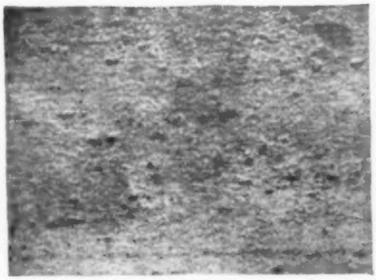
CADMIUM × 30 DIAMS.

Fig. 216.—Rolled and annealed.
" 218.— " " "
" 220.— " " "
" 222.—Annealed and strained.

Fig. 217.—Rolled.
" 219.— " and annealed.
" 221.— " " "
" 223.—Annealed and strained.







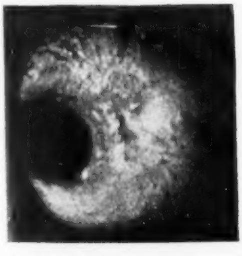
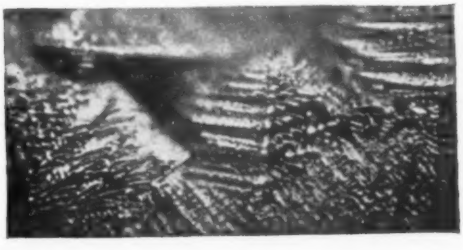



FIG. 224.—Copper, slowly cooled × 30 diams.
" 226.—Cu. impure, cast × 30 "
" 228.—Cu. foil unannealed × 50 "
" 230.—Cu. slowly cooled × 1¼ "

FIG. 225.—Cu. Electrolytic × 50 diams.
" 227.—Rolled Copper × 30 "
" 229.—Cu. foil annealed × 50 "
" 231.—Gold, slowly cooled × 30 "

large amount of spring, but, on heating to a dull red and cooling slowly, this disappeared.

49. Fig. 229, × 50 diameters, shows this same foil after heating to a dull red. All trace of the fine-grained structure has gone, and in its place is a coarser, mottled structure with broad boundaries to the grains.

Gold.

50. The structure of gold has been studied by Arnold and by Andrews.* The latter found that pure gold crystallized in the cubic system: that the primary crystals as seen in section are chiefly hexagonal, and are built up of secondaries which also generally belong to the cubic system. Roberts-Austen and Osmond, in a paper on the structure of metals, its origin and causes, described the structure of gold and the effects of various impurities.

51. Fig. 231, × 30 diameters, shows some of the dendrites met with in a slowly-cooled gold button, and in Fig. 233 is the same under a magnification of about 10 diameters, where the relations between the primaries can be seen.

52. Rolling breaks down the primary crystals, producing a much finer structure, which on annealing grows to a considerable size, as has been shown by Roberts-Austen. The effects of slight strain in producing slip-bands and twinning have been studied by Ewing and Rosenhain, and it would seem that gold closely resembles copper.

Lead.

53. In Fig. 232, is shown a specimen of very pure lead prepared in the laboratory of the late Professor Percy, by pouring away the liquid from a partly solidified crucible of metal. The dendritic structure (reduced $\frac{1}{2}$) is clearly shown.

54. In Fig. 234 are shown three lightly etched ingots of lead. The one on the right has been cast in an iron mould, 5 inches by 0.6 × 0.3 inch; that on the left in a stone mould, 5 inches by 0.6 × 0.2 inch; whilst the centre ingot was cast in a stone mould, but before solidification was complete, the remaining liquid lead was poured off, and in this way the internal structure has been laid bare. When the surface of cast lead is examined, dendrites are seen.

55. Fig. 235, × 20 diameters, oblique illumination, shows such

* "Engineering," 30 September, 28 October, 9 December, 1898.

dendrites. The specimen has been etched with dilute nitric acid, and the dendrites are seen to be the skeletons of the grains or primary crystals. When lead is cast on a flat stone surface better examples are seen, as in Fig. 237, × 34 diameters.

56. In Fig. 238, × 30 diameters, the secondary grains are seen. This surface was the last point to solidify in an ingot cast in stone. Etching brings out the primary crystallization very distinctly.

57. Fig. 239, × 30 diameters, shows the surface of an ingot etched with nitric acid. In addition to the primary grains, another structure is shown up by etching. Within the central grain, small markings are seen; under a high power these are seen to be tetrahedra eaten out by the acid. These tetrahedra are all similarly orientated in the same grain.

58. In Fig. 240, × 30 diameters, is seen the effect of etching the base of an ingot deeply with acetic acid. The primary grains or crystals are well shown, each grain presenting a rough surface made up of tetrahedra with distinct orientation in each grain.

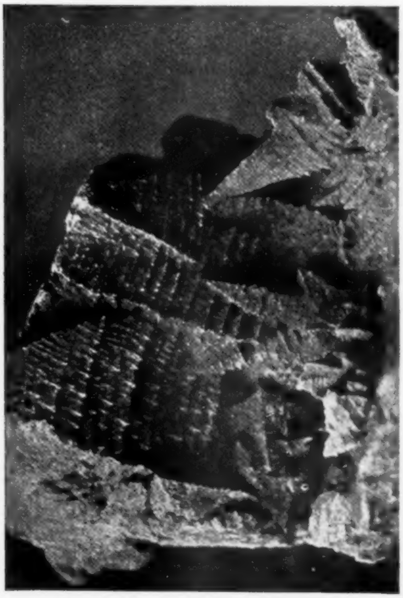
59. Fig. 241, × 30 diameters, shows the surface of an ingot cast in stone, and etched for twenty-four hours with acetic acid, thus showing up the difference in orientation of the secondary grains.

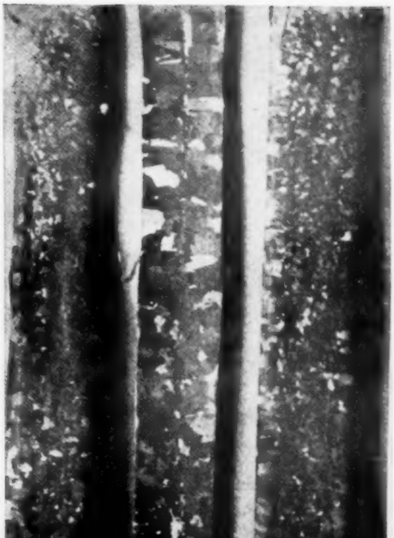
60. Fig. 242, × 30 diameters, shows the base of an ingot cast in stone, and very deeply etched with acetic acid. Orientation here is evidently about a trigonal axis.

61. The effect of strain is to produce innumerable fine slip-lines.

62. Fig. 243, × 35 diameters, shows such lines set up by bending, in lead, cast on a flat stone surface. The boundaries between the primary crystallization have become more pronounced. So far the author has observed no broad slip-bands or twins such as are seen in bismuth, cadmium, etc., but true twins do occur on annealing sheet lead. It was noticed that these strain-lines or lines of slip occur crossing the two main axes of the dendrites at a large angle. Three systems of parallel slip-lines have been observed in a single grain, and these are parallel to the three sides of the tetrahedra eaten out by etching. A fourth possible one would be in the plane of the photograph and parallel to the fourth face of the tetrahedra. Hence it would seem that these slip-lines would coincide with a possible cleavage parallel to the fundamental octahedron (111), if we consider that the two arms of the dendrites coincide with two of the axes of the cube.

63. In Fig. 244, × 30 diameters, a cast surface, etched with





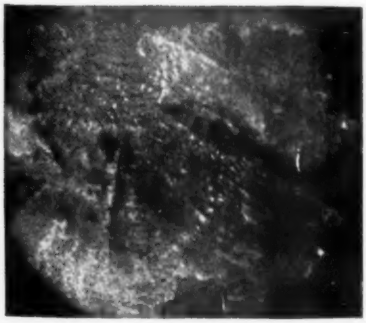
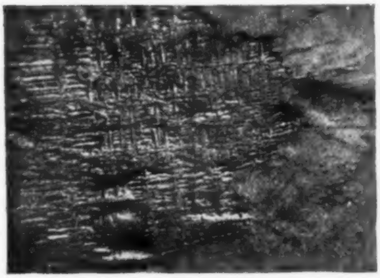
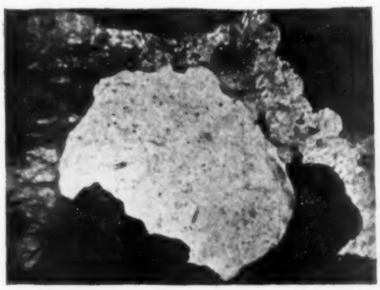
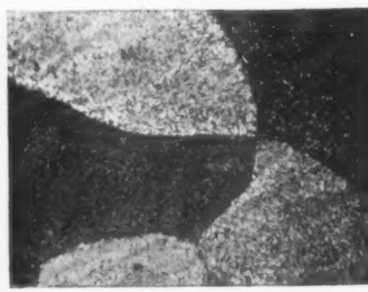
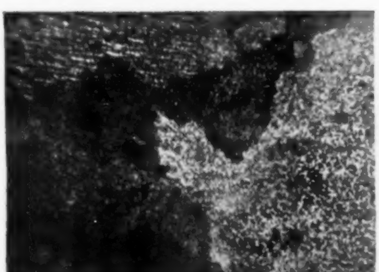
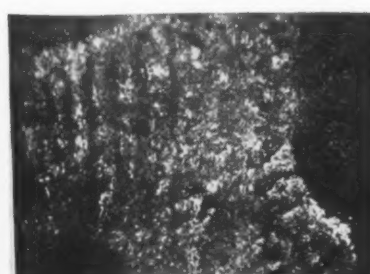
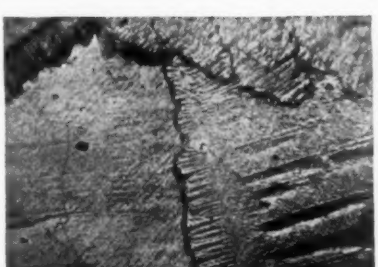


Fig. 232.—Lead crystals (reduced).
" 234.—Lead ingots.

Fig. 233.—Gold slowly cooled × 10 diams.
" 235.—Lead, cast, surface × 20 "
" 236.—Lead, rolled and annealed.




LEAD.

FIG. 237.—Cast × 35 diams.
" 239.— " etched × 30 diams.
" 241.— " " " "
" 243.— " and strained × 35 diams.

Fig. 238.—Cast × 30 diams.
" 240.— " Etched × 30 diams.
" 242.— " " " "
" 244.—Etched and strained × 30 diams.

nitric acid and stained, is seen the relation of the slip-lines to the tetrahedra or secondary crystallization. Hammering and rolling break up the primary crystallization, and a very fine-grained structure results.

64. Fig. 246 shows two strips produced by rolling out small ingots similar to those seen in Fig. 234. They measure 4 inches by 0.6 × 0.12 inch. Under the microscope the structure is similar to that seen in Fig. 217. Annealing causes a rearrangement and the growth of crystals or grains follows.

65. Fig. 248 shows the same pair of strips annealed for seven days at a temperature of about 180 degrees Cent. (356 degrees Fahr.).

66. In Fig. 245 is shown a piece of sheet lead, 3 inches by 2.5 × 0.05 inch thick. It is several years old, and there is no doubt that the difference in size of the crystallization between the freshly rolled metal in Fig. 246 and the old in Fig. 245 is due to growth in the strained metal at normal temperatures, for on rolling out a piece of the old sheet lead a structure as fine as in Fig. 246 was produced.

67. Fig. 247 shows the same sheet annealed for seven days at about 180 degrees Cent. (356 degrees Fahr.), and the growth of crystals is very marked.

68. Fig. 236 shows a piece of sheet lead 2.5 inches by 2 × 0.07 inch annealed for eight days discontinuously. It was annealed during the day, but cooled down to the ordinary temperature of the laboratory during the night. It would seem that discontinuous annealing caused a larger crystallization. The temperature of annealing was about 180 degrees C. (356 degrees Fahr.) as before. Besides several crystals of enormous size compared with the freshly rolled structure, it is noticed that very many of the crystals are twinned. These twins, as in the case of those met with in annealed sheet gold, brass, etc., are therefore the product of annealing. The annealing of lead has recently formed the subject of research by Ewing and Rosenhain,* who have even made crystals cross a weld by annealing at 200 degrees C.

Platinum.

69. Fig. 249, × 30 diameters, shows the dendrites found on the surface of a platinum button. They consist of two axes at right angles, and form the skeletons of the grains or crystals as

* Philosophical Transactions, Royal Society.

in the case of the other metals examined. Platinum has the same granular structure as gold, etc., secondary grains being revealed in the large crystals by deep etching, whilst strain produces slip-lines related to these etching pits. Andrews has recently worked out the structure of platinum.*

Silver.

70. Fig. 250, × 30 diameters, shows the surface structure of an ingot of silver. Three or more primary crystals or grains are seen to be built up of numerous secondaries, possessing distinct orientation.

71. Fig. 251, × 15 diameters, shows the surface of a button allowed to cool slowly in the crucible under a cover of borax. The crystallization is extremely coarse, the illustration showing only part of four parallel branches of a single dendrite. The difference between Figs. 250 and 251 is due to the difference in rate of cooling.

72. Fig. 252, × 30 diameters, represents the surface of a small ingot cast under a thick cover of salt. There is a very curious dendrite structure, which in places passes into coarse grains as in Fig. 253, × 30 diameters.

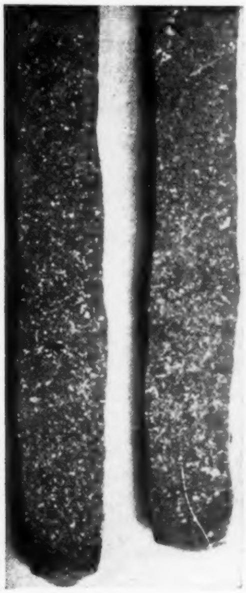
73. In Fig. 255, × 30 diameters, is seen the intermediate stage between Figs. 252 and 253.

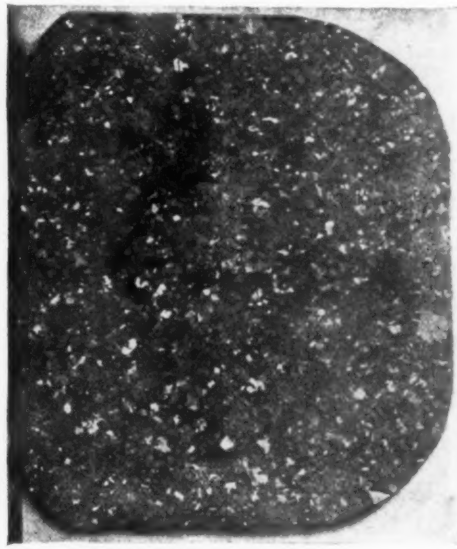
74. Fig. 254, × 8 diameters, shows the structure of electrolytically deposited silver. Distinct crystalline forms are seen, and Fig. 256, × 8 diameters, shows some of the simpler variety. It was attempted to cut sections of the more massive growth to obtain their internal structure, as in the case of copper, but without success. Electrolytic iron and nickel, on the other hand, form a deposit consisting of numerous coatings like an onion, and a section shows that structure. If a cupellation button of silver be examined very good examples of dendritic structure are often seen. Figs. 257 to 262 are all taken from a single button.

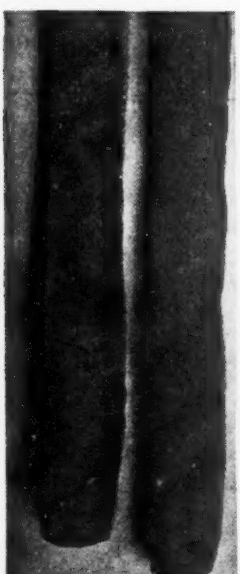
75. In Fig. 257, × 30 diameters, is seen the centre of crystallization of a dendrite. Two axes are seen at right angles, and from each axis there is a perpendicular growth, growing coarser as we leave the centre of crystallization. There is a cubic symmetry.

76. Fig. 259, × 30 diameters, shows one of the arms or axes near its centre, whilst Fig. 261 shows the end of the same. As the

* Philosophical Transactions, Royal Society.







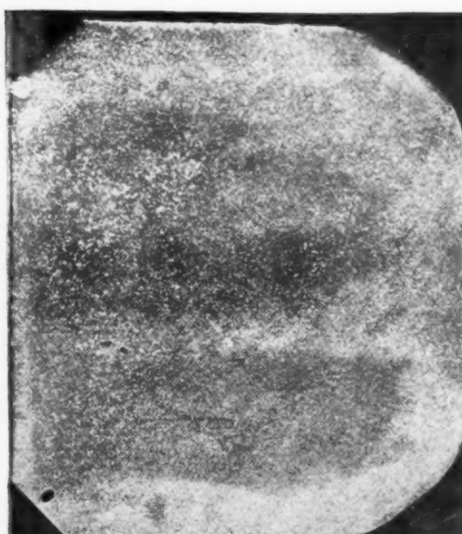
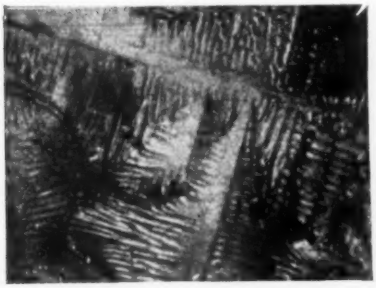
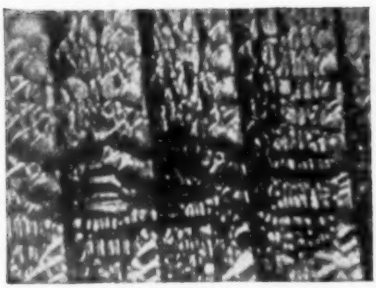
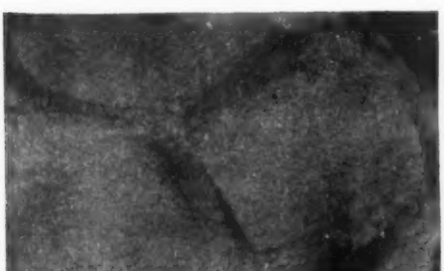


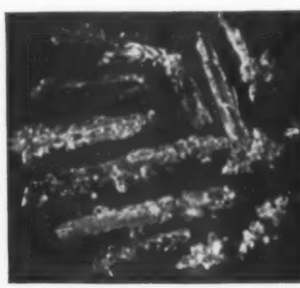
FIG. 245.—Sheet lead.
" 247.— " " (Fig. 245 annealed).

FIG. 246.—Rolled lead.
" 248.— " " (Fig. 246 annealed).









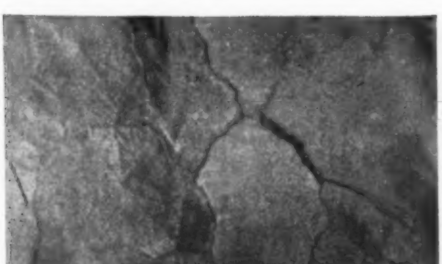


FIG. 249.—Platinum slowly cooled × 30 diams.
" 251.—Ag. slowly cooled × 15 diams.
" 253.— " cast under cover of salt × 30 diams.
" 255.— " " " " " " "

FIG. 250.—Silver ingot, surface × 30 diams.
" 252.—Ag. cast under salt × 30 "
" 254.—Ag. electrolytic × 8 "
" 256.— " " × 8 "

secondary grains approach the end of the axis they themselves possess a structure in which there are two main axes, together with a radial as well as a concentric structure.

77. Fig. 258, ×30 diameters, shows the junction of two primary grains or crystals, in which the radial and concentric structure of the secondary grains is very marked. A marking similar to the concentric one is seen at the end of an axis, as in Fig. 261, × 30 diameters.

78. Figs. 260 and 262, × 15 and 30 diameters respectively, oblique illumination, show the normal structure of the surface. A few of the grains or crystals, as in Fig. 262, show a radial as well as a concentric structure. At first it was thought that these marked polygonal boundaries were those of the primary crystals, but on closer examination they are seen to cut across the grains in several places. These pseudo-boundaries, it is true, do coincide with the real boundaries, as a rule, but in many cases they are seen to be distinct. As Ewing and Rosenhain have shown, the difference between the two is clearly shown by slip-lines when the metal is strained. The pseudo-boundaries are the result of contraction in cooling, like the columnar structure of basalt.

79. A silver cupellation button, from which the copper had not been entirely removed, was examined. Although the axial arrangement of the secondary grains in the primary was not so sharp, when present, as in the case of pure silver, the concentric and radial structure of the smaller primaries was more marked.

80. Fig. 263, × 15 diameters, shows the axial arrangement of a dendrite. The extremity of an axis is seen in Fig. 264, × 15 diameters, oblique illumination, together with some of the smaller primaries. It is very hard to distinguish between the largest secondary grains of a dendrite and the smallest primaries, and it is probable that, as they are of the same generation, the one set passes into the other.

81. Figs. 265 and 266, ×30 diameters, show the radial markings and the boundaries of the large grains. The distinct boundaries may be due to a trace of eutectic present. When an ingot or button of silver is strained—as, for instance, by hammering—marked parallel twinning is seen. The broadest twins usually lie perpendicular to the line of strain. In addition to these broad bands are numerous series of finer parallel twinning, running at all angles. In some cases the straining appears to have produced

Tin.

82. When an ingot of tin is cast, it possesses a plain bright surface if pure. If impure, the surface is covered more or less by dendritic crystals due to the formation of mixed crystals, whose outer portions solidify at a slightly lower temperature than their centres. In the case where a eutectic is formed, this shrinks on cooling, leaving the dendrites standing above the surface. If the surfaces of ingots be etched, they are seen to be coarsely granular, the size of the grains, and to some extent their shape, varying with the mould used and with the purity of the metal. The temperature of the mould and of casting also alters the size of the grains, the higher the temperature, the larger the grains.

83. Fig. 267 shows four ingots of tin which have been etched with dilute nitric acid and then cleaned with dilute hydrochloric, which removes the oxide of tin very quickly.

a is pure tin cast in a cold iron mould. Its dimensions are 5 inches by 0.6 × 0.4 inch thick.

b is pure tin cast in a stone mould. Dimensions are 5 inches by 0.6 × 0.2 inch.

c is pure tin cast in an elliptical iron mould. Its largest dimensions are 4.5 inches by 0.7 × 0.4 inch.

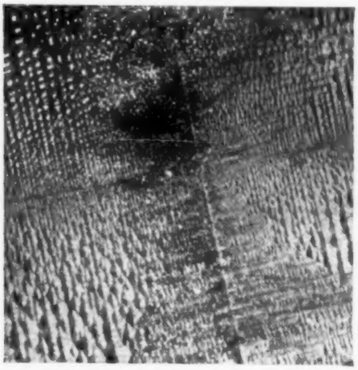
d is impure tin cast in the same mould as *a*, but is only 0.3 inch thick.

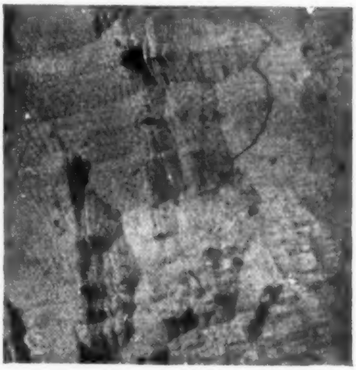
84. The structures of *a*, *b*, and *c* are all similar, but that of *d* is of a much finer character, the grains being much more elongated, and point towards the centre of the ingot. If, in casting an ingot, the mould be tilted and the liquid tin poured away just before solidification is complete, a difference in structure is noticed between pure and impure tin.

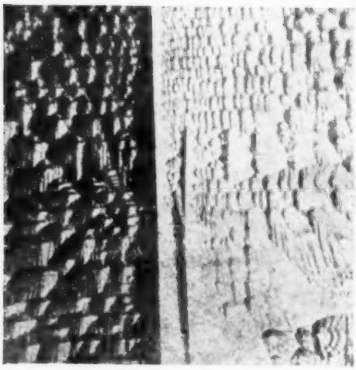
85. Fig. 280 shows two such ingots—one pure, the other impure. That on the left is very impure, and the whole mass is seen to be composed of dendrites and skeleton crystals. That on the right is pure grain tin and no dendrites can be seen.

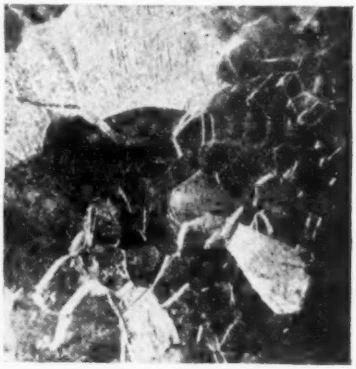
86. Fig. 268, × 30 diameters, shows the dendritic structure of tin, which has been cast on a flat stone surface. Part of the main axis of a dendrite is seen.

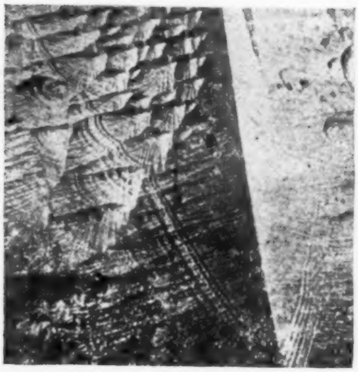
87. In Fig. 269, × 30 diameters, is seen the base of a very thin sheet cast on stone and a further development of the structure appears. This shows the junction of two grains with their dendritic framework.

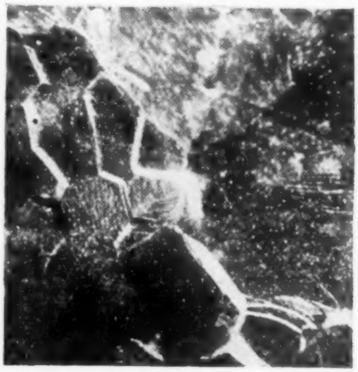








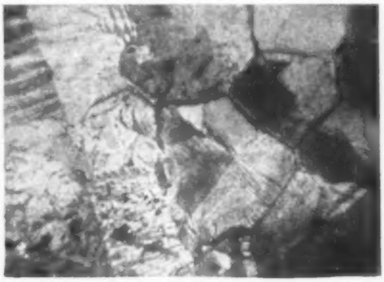


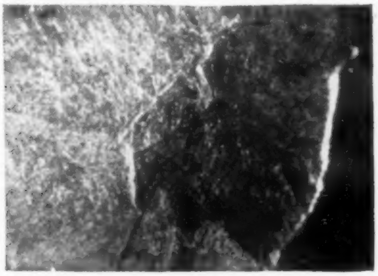


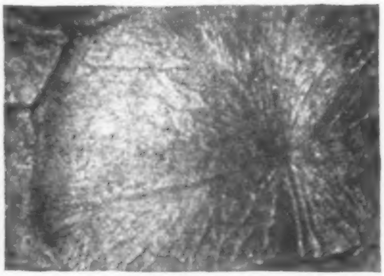
SILVER.

FIG. 257.—Cupelled × 30 diams.
" 259.—Continuation of Fig. 57.
" 261.— " " 59.

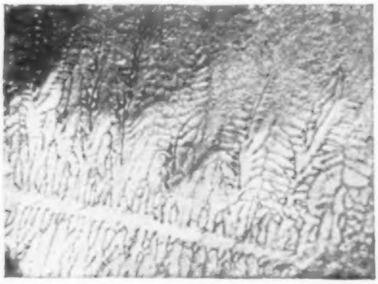
FIG. 258.—Cupelled × 30 diams.
" 260.— " × 15 "
" 262.— " × 30 "







d c b c



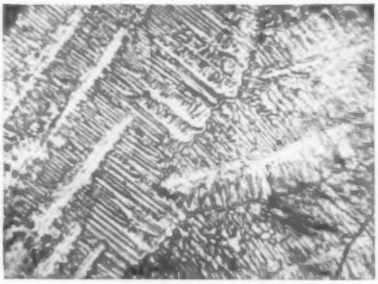
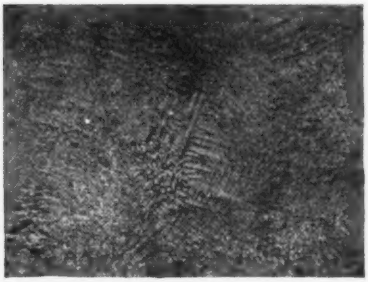
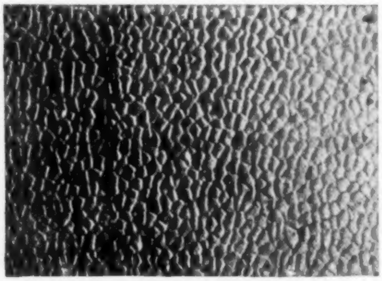
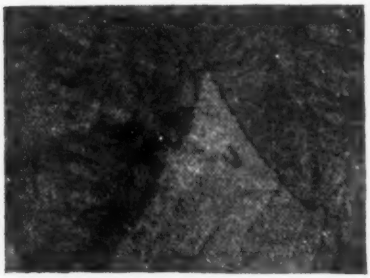


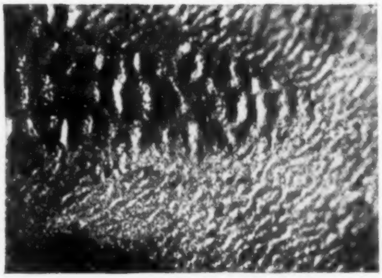
FIG. 263.—Ag. cupelled containing cu. × 15 diams.
FIG. 265.—Ag. cupelled containing cu. × 30 diams.
FIG. 267.—Ingots of tin.

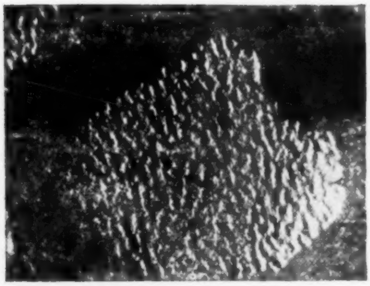
FIG. 264.—Ag. cupelled containing cu. × 15 diams.
FIG. 266.—Ag. cupelled containing cu. × 30 diams.
FIG. 268.—Tin cast on stone × 30 diams.
" 269.— " " " " × 30 "

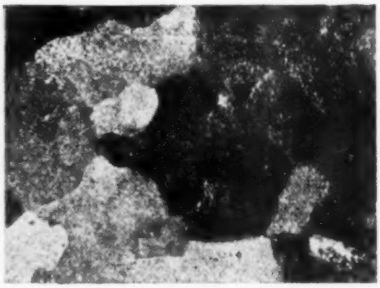


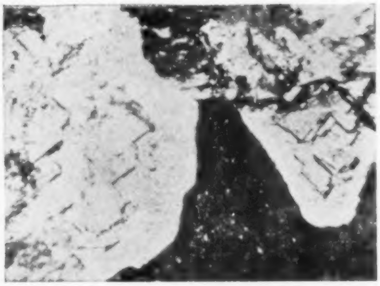








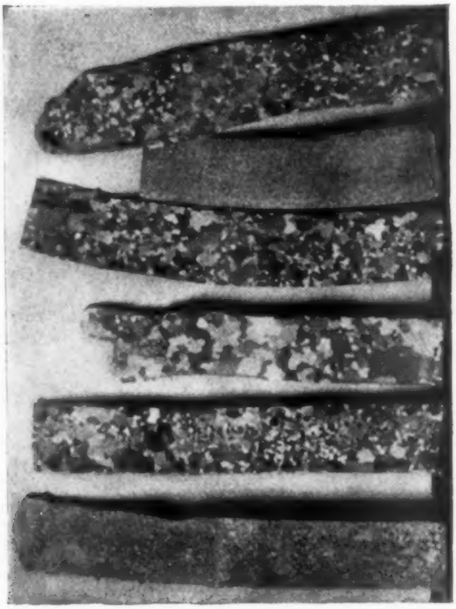


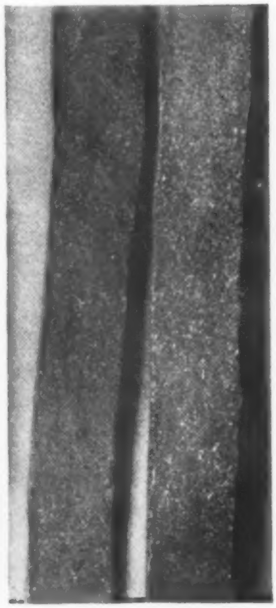


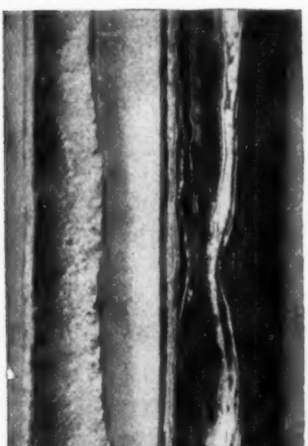
TIN.

FIG. 270.—Cast, surface × 30 diams.
" 272.—Dendrites × 30 "
" 274.—Cast, deeply etched × 30 "
" 276.— " slowly " × 30 "

FIG. 271.—Cast surface × 75 diams.
" 273.—Denerites deeply etched × 30 diams.
" 275.—Cast, etched × 30 "
" 277.— " slowly etched × 30 "



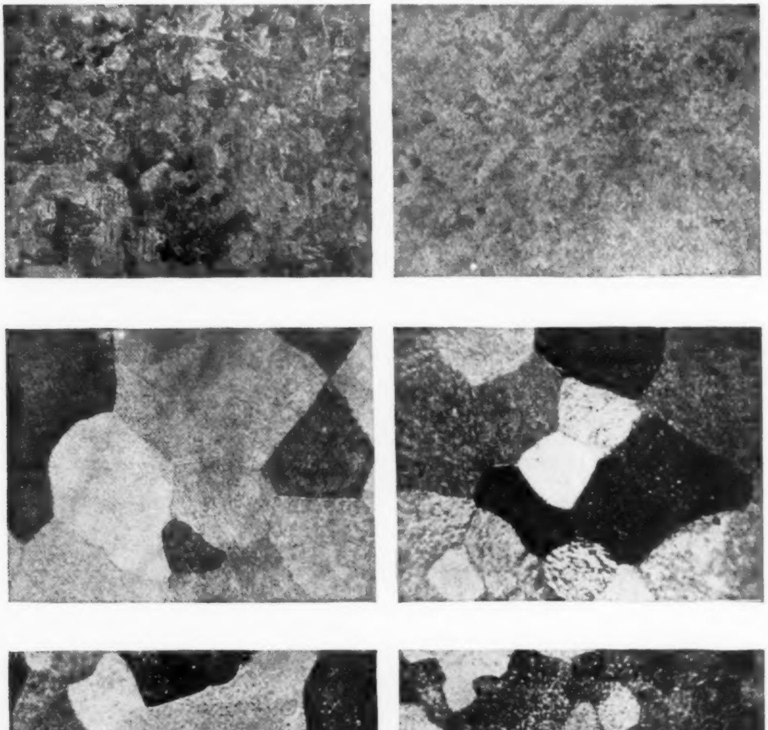




TIN.

FIG. 278.—Rolled and annealed.
" 280.—Cast.

FIG. 279.—Rolled.
" 281.—Annealed (Fig 79).



TIN.

FIG. 282.—Rolled 0.5 mm thick × 30 diams.
" 284.—Annealed 0.9 mm " × 15 "
" 286.— " 0.5 " " × 15 "
" 288.—Hammered × 30 "

FIG. 283.—Rolled 0.1 mm thick × 30 diams.
" 285.—Annealed 0.9 mm thick × 15 diams.
" 287.— " 0.25 " " × 15 "
" 289.—Hammered and annealed × 35 "

88. Fig. 270, $\times$ 30 diameters, shows a smaller dendrite from the surface, with two main axes at right angles, set in a groundmass which is made up of small secondary grains. These secondary grains of tin are well shown in Fig. 271, $\times$ 75 diameters. When pure tin is cast in a mould, especially in an iron one, wherever it has solidified last a concentric arrangement due to shrinkage is seen. It is generally extremely slight, but may be accentuated by giving the mould a slight jar, when vibrations are set up. This structure is seen at the lower part of ingot *c* in Fig. 267. It also occurs in slightly impure tin, as at the centre of the surface of ingot *d*. If the centre of this point of last solidification be examined under the microscope, the normal structure of the secondary grains is seen, as in Fig. 271.

89. Fig. 272, $\times$ 30 diameters, oblique illumination, shows leaf-like dendrites often found on the surface of ingots. In this case, as in all dendrites found at the surface, the secondary grains are much larger than those of the surrounding tin, and are more regularly formed, due to the fact that they had more freedom of growth.

90. Fig. 273, $\times$ 30 diameters oblique illumination, shows dendrites similar to those in Fig. 272, after very deep etching with dilute nitric acid.

91. Fig. 274, $\times$ 30 diameters, shows the normal surface of cast tin, etched for twelve hours with dilute nitric acid. The large primary crystals or grains, built up of small secondaries with definite orientation, are clearly seen.

92. Fig. 275, $\times$ 30 diameters, shows the base of an ingot etched for twelve hours. The primary and the secondary grains are much smaller, due to their having solidified against the comparatively cold surface of the iron mould.

93. In Figs. 276 and 277, $\times$ 30 diameters, oblique illumination, are seen the large grains of an ingot of pure tin, which has been etched for 24 hours with very dilute hydrochloric acid to ensure very slow action. The secondary grains show plane faces, and closely resemble cleavage as seen in minerals. In some cases pits were eaten out, having plane faces with the form of an octahedron. Ewing and Rosenhain have obtained a characteristic structure by casting against glass.

94. When tin, either cast or slowly cooled, is rolled, the original structure is destroyed, and a new very fine crystallization is set up.

95. Fig. 279 shows a bar of tin cast in stone rolled out.

Its dimensions are 4 inches by 0.7 inch, thickness 0.126 inch. Its structure has been entirely destroyed, and this fine crystallization has been set up in its place. On annealing, rearrangement takes place, and a structure consisting of coarse grains is produced.

96. Fig. 281 is part of the same rolled bar, after annealing for ten days at a temperature under 200 degrees C. (392 degrees Fahr.). Large grains have grown out of the fine crystallization, their orientation is marked, and their boundaries distinct.

97. In order to find out whether any relation existed between the thickness of the rolled bar and the size of the grains, unannealed and annealed, one of the bars, Fig. 279, was put through the rolls till it became reduced in thickness to 0.035 inch. Half of it was cut into lengths from 2.9 to 2.2 inches long and 0.5 inch wide. The other half was rolled down to 0.02 inch and cut into lengths of about 3 inches (0.5 to 0.6 wide). One of these was rolled down to 0.01 inch, another was taken to foil 0.004 inch thick. Samples of each were annealed for five days on a hot-plate at a temperature of 180 degrees C. On etching it was found that the size of grains in all the annealed specimens varied considerably, small and large being found in each. But it was seen that the thicker the metal the more numerous were the large grains.

98. Fig. 278 shows six of the strips referred to above. The first three pieces are 0.035 inch thick, the first of which is the unannealed specimen. The last three are 0.02 inch thick, and the centre one is unannealed. The contrast is marked.

99. Figs. 298 and 299 show two pieces 0.01 inch thick, which have been annealed with the others, but, after annealing, one end of each was quickly raised to the melting point by means of a Bunsen beneath the hot-plate. As soon as the tin started to melt, the Bunsen was turned out and the whole slowly cooled. The junction of the annealed and remelted parts is clearly seen. In the former, there are the usual grains more or less equi-dimensional, whilst in the latter it is seen that a few crystals occupy the whole mass. It is seen that the large long crystals of the melted part have grown from and are continuous with the crystals in the unmelted part. This shows the relative freedom to grow in the liquid and in the solid states.

100. Under the microscope the difference between the rolled metal of various thicknesses is distinct. The piece 0.035 inch thick showed a structure as coarse as in Fig. 287, but the grains had rough irregular boundaries.

101. Fig. 282, × 30 diameters, oblique illumination, shows the structure of the piece 0.02 inch thick. The grains have become minute and are less distinct.

102. Fig. 283, × 30 diameters, oblique illumination, is the piece rolled down to 0.004 inch. Its structure is minute but granular, and the grains are seen to be rather more distinct than those of the 0.02 inch strip. In the case of the annealed specimens, the difference between the pieces of 0.035 and 0.02 inch in thickness is not very noticeable, but when 0.01 inch is reached the grains are seen to be very much smaller.

103. Fig. 284, × 15 diameters, oblique illumination, shows the piece 0.035 inch thick after annealing. The grains or crystals have increased many times in size, their edges have become more regular, and the whole system of crystallization has become more distinct.

104. Fig. 285, × 15 diameters, oblique illumination, shows some of the smaller crystals from the same strip.

105. Fig. 286, × 15 diameters, oblique illumination, shows the 0.02 inch strip after annealing, the size of grain being but slightly, if any, less than that of the thicker strip.

106. In Fig. 287, × 15 diameters, oblique, is seen the 0.01 inch strip after annealing. The grains or crystals are very much smaller. On comparing Fig. 282 with Fig. 286, and Fig. 283 with Fig. 287 the enormous change due to annealing is clearly seen, and this is even more marked because the annealed strips have been photographed with only half the magnification of the rolled specimens.

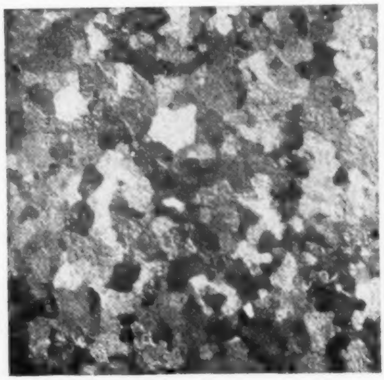
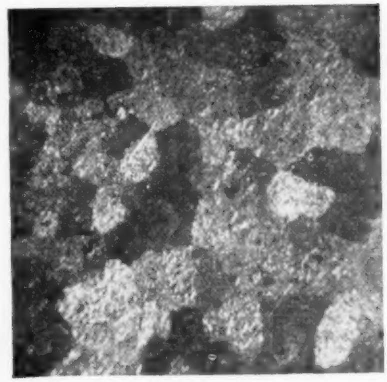
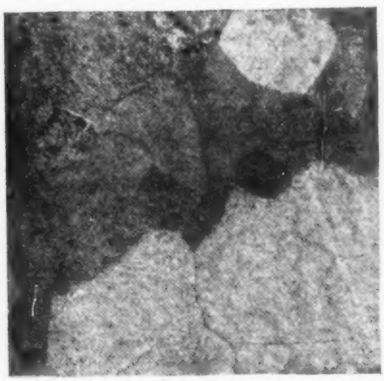
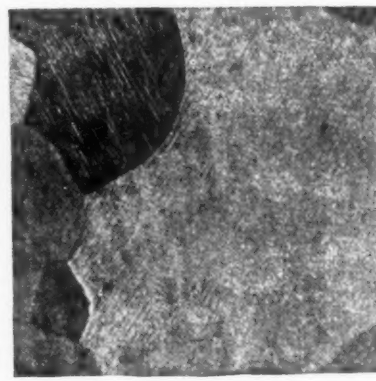
107. Hammering produces similar effects to rolling.

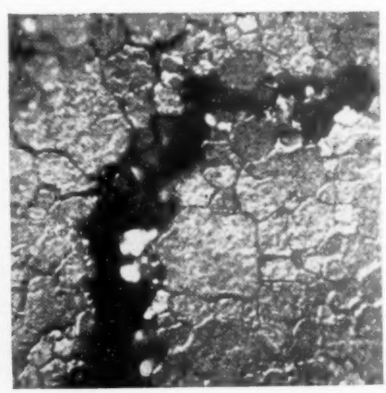
Fig. 288, × 30 diameters, oblique illumination, shows a piece of tin hammered out to 0.08 inch thick, etched with dilute nitric acid and washed with dilute hydrochloric. The original structure has disappeared, and a finer crystallization has taken its place. Annealing causes a growth of crystals as before.

108. Fig. 289, × 35 diameters, oblique, shows the effect of annealing for three hours at a temperature below 200 degrees C. (392 degrees F.), whilst in Fig. 290, × 35 diameters, is seen a patch of the largest crystals grown in that time.

109. In Fig. 291, × 33 diameters, is shown a piece of the same tin hammered to 0.05 inch thick, and annealed for fifteen hours, showing a great increase in size and regularity of crystals.

110. Fig. 292, × 30 diameters, shows the structure of ham-



TIN.

FIG. 290.—Annealed × 35 diams.
" 292.— " 10 days × 30 diams.
" 294.—Heated to melting point × 35 diams.

FIG. 291.—Annealed × 33 diams.
" 293.—Horiz. sect. of (Fig. 92) × 30 diams.
" 295.—Fracture of (Fig. 94) × 35 "

mered tin after annealing below 200 degrees C. (392 degrees F.) for ten days. When compared with Fig. 288, the difference in structure due to annealing is very striking. It would seem that the fine crystals tend to become more definite in boundary, their several neighboring grains become oriented in the same way, and finally merge into one another to form a large grain or crystal. The largest grain in Fig. 292 shows the outline of the smaller grains which build it up; their outline is the result of etching after a slight annealing.

111. Fig. 293, $\times$ 30 diameters, shows a section of the same. Curious rectilinear markings are seen. These are not scratches, and are distinct from the slip-bands and lines produced by straining. They are differently oriented in different crystals. They may however have been produced by strain in the process of cutting the section.

112. Fig. 294, $\times$ 35 diameters, oblique illumination, shows a piece of hammered tin 0.08 inch thick which has been slowly annealed up to the beginning of fusion. It was supported at each end and heated until it broke by its own weight. The fracture is seen in Fig. 295, $\times$ 35 diameters. The coarse granular structure produced by the heating has been shown up by fusion at the boundaries.

113. Tschernoff, who is followed by Sauveur, says that the growth or coarsing of the grain of steel as revealed by its fracture cannot occur during rise of temperature, but occurs only during fall of temperature. This may be true for the constituent ferrite, but the growth of grain of the martensite takes place as the temperature rises (above Ac) just as in the above example of tin.

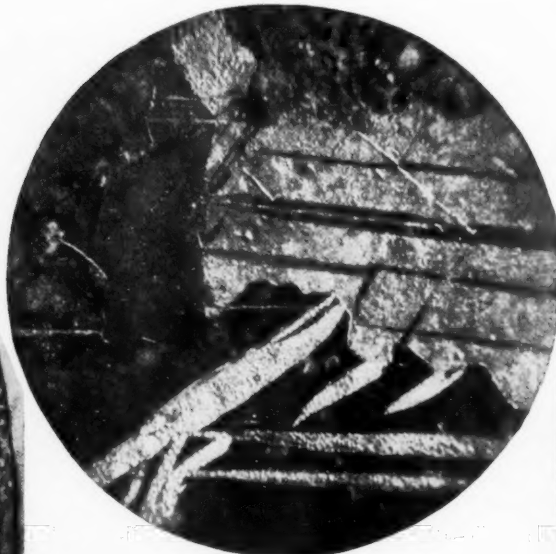
114. When tin is strained by bending, slip-bands or twins are produced as well as slip-lines. Figs. 296 and 297, $\times$ 30 diameters, show such twinning produced in cast tin (cast on a flat stone surface). The metal has been etched with dilute nitric acid. In Fig. 297 the two parallel systems are very pronounced.

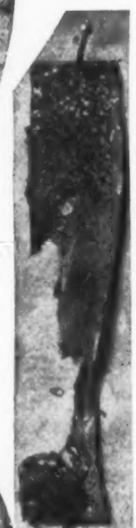
Zinc.

115. When an ingot of zinc is examined, dendrites similar to those of lead and tin are often found.

Fig. 303, slightly reduced, shows the surface of an ingot cast in a stone mould. The granular crystallization is shown in Fig. 302 by etching.

Fig. 304, $\times$ 30 diameters, shows part of a dendrite on the





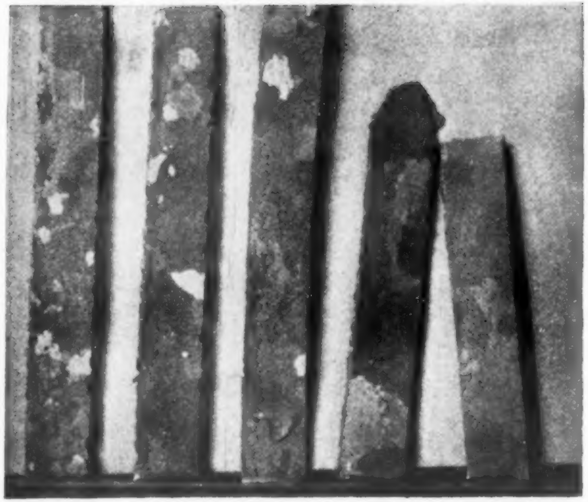
TIN.

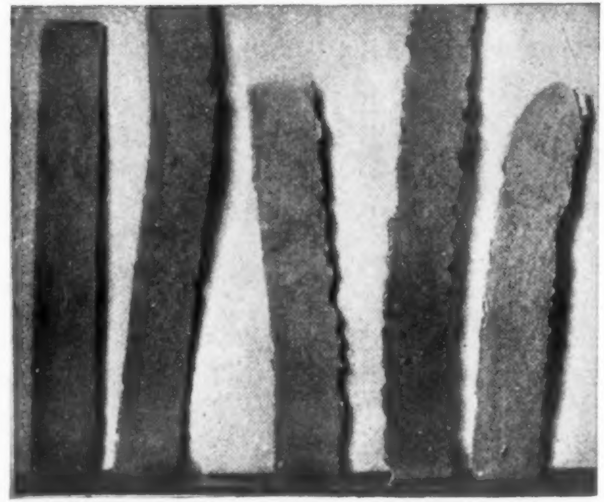
FIG. 296.—Strained and etched × 30 diams.

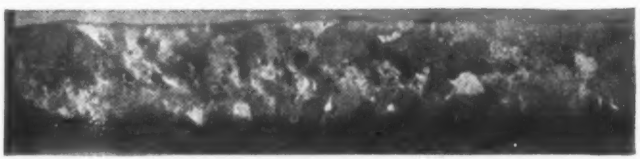
" 297.— " " " × 30 "

FIG. 298.

FIG. 299.







ZINC.

FIG. 300.—Rolled and Annealed.
" 302.—Ingot, etched.

FIG. 301.—Rolled.
" 303.—Ingot, surface.

surface of metal cast on stone. There are three main axes, giving the dendrite a hexagonal type of structure.

Fig. 305, × 30 diameters, shows the base of the same metal with its coarse granular structure. Etching reveals the difference in orientation of the secondary grains as in the case of lead, tin, etc.

Straining produces parallel twinning as before.

Fig. 306, × 30 diameters, shows this structure in a surface similar to Fig. 305.

In Fig. 307, × 30 diameters, several twinnings are seen, which vary in direction as they pass from one grain to another. Etching makes the effect more pronounced by showing up the orientation of the grains as in Fig. 308, × 30 diameters, oblique illumination.

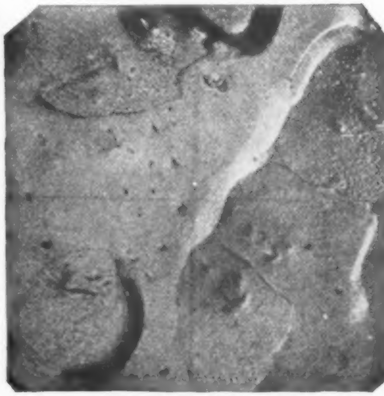
116. In Fig. 309, × 30 diameters, is illustrated a view of the surface of the same metal. The primary grains are coarser, due to the relatively slower cooling, and the twinning is more pronounced. When zinc is rolled out into thin strips, the metal is very pliable. The structure of the original cooling has been entirely destroyed, and a new one takes its place, much finer than that of rolled tin, lead or cadmium.

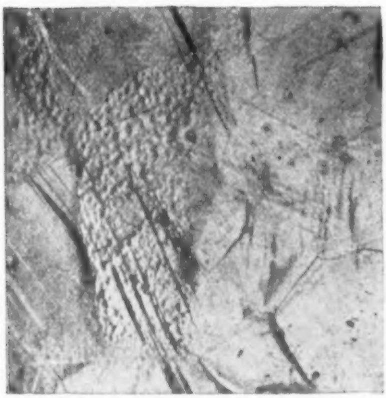
Fig. 301 shows five strips rolled out to 2 to 4 by 0.4 by 0.04 inch thick. The structure cannot be seen without the aid of a lens.

117. Fig. 300 shows the same strips annealed for seven days at 200 degrees C. (392 degrees F.) on a hot plate. Large coarse irregular grains or crystals have been formed, far larger than in the case of tin, lead, etc. Whereas the unannealed strips are flexible and very tough, the annealed specimens are very brittle. Bending or straining to the slightest degree causes the development of cracks which originate as slip-bands more or less parallel to the normal of the line of strain. This brittleness is so pronounced that the metal cannot be rolled until it is heated to its critical temperature. Experiments seem to show that discontinuous annealing causes a more rapid growth than a continuous one. In other words, several annealings of short duration cause the crystals to grow in less time than one long annealing.

Summary.

118. In the above examples is given additional evidence of what has been pointed out by others, that when a metal reaches its freezing point it begins to crystallize out from a number of points or centres. Dendrites grow from these centres and continue to





ZINC.

Fig. 304.—Cast, surface × 30 diams.
" 306.—Cast, strained × 30 "
" 308.—Strained, etched × 30 "

Fig. 305.—Cast base × 30 diams.
" 307.—Cast, strained × 30 "
" 309.—Strained, etched × 30 "

grow until they meet others, when their growth is obstructed. The more rapidly a metal cools past its freezing point, the more numerous will be the centres of crystallization, and therefore the smaller will be the structure. The dendrites continue to grow until the whole mass becomes solid in the form of irregularly bounded crystals or grains whose orientation varies from grain to grain. In the case of impure metals the dendrites which first crystallize out are usually purer than the mother-liquor, which freezes at a lower temperature. As a rule equilibrium is not established, and so the dendrites vary in composition from centre to outside, hence their structure is revealed when a section is etched. Again, as a rule, the metal contracts during solidification, and so the mother-liquor sinks beneath the surface, leaving the dendrites standing out in slight relief. These primary grains or crystals are built up of smaller or secondary grains which are revealed in etching. They may be compared with the crystals of calcite in a crystalline limestone which break up into small rhombs due to their rhombohedral cleavage, or with the crystals in massive galena which break up into cubes because of their cubic cleavage.

119. The effect of strain is to produce a parallel slipping along definite directions, usually related to the orientation of the primary crystals and also to the direction of strain. In some cases the effect is seen in the formation of one or more systems of parallel slip-lines, in others in the production of systems of parallel twinning. As Ewing and Rosenhain have pointed out, the first slip-lines are perpendicular to the direction of straining, whereas further straining produces other systems. When the strain has been severe, as, for instance, by rolling or hammering, the large primary crystals are broken down and a finer crystallization takes their place; the greater the mechanical work upon the metal, the finer the resulting crystallization.

120. Annealing causes the fine crystallization to rearrange itself, and produces the growth of crystals, whose size apparently depends to some extent both on the time and temperature of annealing and the thickness of the metal. In the case of lead and tin this arrangement was noticed to take place at ordinary temperatures, but of course very slowly indeed.

In conclusion the author would like to express his indebtedness to the late Sir William Roberts-Austen, under whose direction most of the work was performed, and also to Professor William Gowland for his kindly encouragement and advice.

No. 1035.*

EXPERIMENTS WITH A LATHE-TOOL DYNAMOMETER.

BY J. T. NICOLSON, D.SC., PROFESSOR OF MECHANICAL ENGINEERING AT THE MUNICIPAL SCHOOL OF TECHNOLOGY, MANCHESTER; SOMETIME PROFESSOR AT M'GILL UNIVERSITY, MONTREAL.

(Member Institution of Mechanical Engineers.)

1. In the tool-steel trials made by the Manchester Committee in 1902-03 (the report upon which was published by the Manchester Association of Engineers in their Transactions for 1903), there appeared an entire lack of uniformity in the shapes and angles of the tools submitted by the eight competing firms. There was also no obvious connection between the shapes and angles of the tools and the cutting forces upon these tools deduced in the report from the electrical power measurements made by the Committee. Neither did the shape or angle supply a clew to the causes of success and failure in the various trials with different tools.

2. On the other hand, the necessary reconsideration of the design of lathes for the rapid and heavy cutting rendered possible by the new steels introduced by Taylor and White, and now everywhere adopted, calls for a thorough and systematic investigation of the forces acting upon a cutting tool. If a standard area of cut can be agreed upon for the various sizes of lathe, a knowledge of the forces to be overcome when taking that cut,—not only for turning the work against the tool, but also for moving the slide-rest and saddle in both the traversing and surfacing directions will enable the calculation of the stresses in, and the proportioning of, the various parts of the machine to be gone about in a rational and scientific way.

3. No such knowledge has hitherto been available; and it appeared to the author that the prosecution of a somewhat extensive

---

* Presented at the Chicago meeting, May and June, 1904, of the American Society of Mechanical Engineers, and forming part of Volume XXV. of the *Transactions*.

research into the matter would well repay the time, labor, and expense, which it would necessarily involve.

4. The present Paper records the results of over 300 serial trials, each requiring the making, recording and reducing of from 50 to 100 observations; but it is only to be looked upon as a first instalment of the work required to be done in order that the action of the tools used in a machine-shop may be thoroughly understood.

5. The thanks of the author, and, if the results herein contained should prove of value, those of the engineering public, should be given to those whose action alone rendered possible the carrying out of the work. First, viz., to the authorities of the Manchester Municipal School of Technology, who authorized the expenditure incurred for power, light, and mechanical assistance, to a not inconsiderable amount. Second, to the firm of Sir W. G. Armstrong, Whitworth & Co., for the continuance of their loan of the lathe used in the experiments by the Manchester Committee; for the donation of the remainder of the material unoperated upon in these experiments, viz., three steel forgings and three iron castings; and for the gift of large quantities of their AW steel of various sections.

6. The record of the experimenters in this field is not a very long one. In Hartig's work "Experiments on the Efficiency of Machine-Tools,"* the law was enunciated that the cutting force varies in simple proportion to the depth of the shaving. In the "Proceedings of the Royal Society" for December, 1881, Mr. A. Mallock published certain observations, and the conclusions he deduced from these, upon lathe turnings, made in the engineering workshop at Cambridge University. He then gave an analysis of the forces acting upon the tool when removing the shaving; and it will be interesting to compare his results, where possible, with the experimental data given below. Professor R. H. Smith published, in his work on "Cutting Tools" in 1882, a series of diagrams and tables, giving the results of experiments made by him with a lever form of dynamometer for measuring the vertical force only; but the cutting forces he measured never exceeded 1,000 pounds, and the cutting speeds were all below 10 feet per minute.

7. Professor Smith, in the preface to his book, takes Hartig severely to task for proposing the above-mentioned law of varia-

* Leipzig, 1873.

tion of force with depth; but (although the author has not had access to Hartig's work) the results of the present experiments appear to substantiate Hartig's law to, at all events, a first approximation; and, however valuable Professor Smith's work may be, it appears to the author to be open to the criticism of lack of scope and exaggeration of detail. His experimental apparatus, consisting of a Smith and Coventry's tool-holder, with a $\frac{5}{16}$-inch steel pin driven through it to act as a fulcrum between the cutting force at the front and the weight scale at the back, was obviously unsuited for heavy cuts, the steel pins having to sustain the sum of both the cutting and weighing forces. In the dynamometers described in this Paper this error has been avoided, and by placing the horizontal and vertical axis about which the tool is free to move at the back, whilst the weighing thrust is applied at the front near the cutting end, and therefore between point and axis, the latter has only the small difference between the cutting and weighing forces to sustain and can be made correspondingly small and frictionless.

8. Two dynamometers were made and used in these experiments. They were each capable of measuring forces up to 15 tons on the tool point when taking a cut. In the first, means were only provided for measuring the vertical force upon the point of the tool; whilst in the second, not only the vertically directed force, but also that tending to push the tool and saddle backwards, and that tending to oppose the traversing feed were observed.

9. The second apparatus grew out of the first, and was only constructed after sufficient experience with the simpler form had indicated the feasibility of a still freer suspension of the tool.

10. The force measurer itself consisted of an hydraulic support and a Bourdon gauge. The author had already had considerable experience with these supports, having constructed small ones for similar work, and a set for use in a rotatory transmission dynamometer, in the laboratory of the McGill University, Montreal. His attention was first attracted to them in connection with the reports on railway brakes presented to this Institution by Captain Galton,* in which a support designed by Mr. George Westinghouse is figured and described, and was largely used in the experimental van employed by the former. The pressure was, however, recorded by connecting the interior of the support

* *Proceedings*, 1878, pp. 467 and 590, also 1879, p. 170.

to a steam-engine indicator, and as there was necessarily considerable leakage of the fluid past the piston, an auxiliary fluid-supply device had to be added, which not only disturbed the readings of the pressure, but also made the instrument more complicated.

11. By reading the pressure on a Bourdon gauge, and taking precautions in making the joints, leakage can however be entirely eliminated; and the author used the instrument in this form in 1894, believing the method to be original. He has since found that many others had adopted the same plan for measuring a transmitted force, notably Napoli, Thomasset, and Maillard.

12. The principal hydraulic support, used for measuring the vertical force on the tool in both the first and second dynamometers, is shown in sectional elevation in Fig. 313. The compression piece *A*, when thrust upwards, produces pressure in the fluid, which communicates by means of the small copper pipe *B* through the make-up plug fitting *C* with the interior of the gauge tube *D*. The body of the diaphragm casing *E* is held down to the lathe saddle by means of two bolts *FF*, Fig. 312. Care must be taken in filling the diaphragm to remove all air from the contained fluid. Distilled water, which is boiled after filling the casing, makes a satisfactory medium; and a small filling screw fitted in the end of the gauge tube is necessary for ridding it of air.

13. Turning now to the other end of the dynamometer, the tool was, in the first instance, free to move only about a horizontal axis (parallel to the lathe centre line). This arrangement is shown in Figs. 310, 311, 312. The axis is formed by the points of two screws *G* passing through two massive cast-iron chocks *H* which rest upon the tool-slide. The points of the screws enter into deep centre pops made on the sides of the tool before hardening. When the tool is resting freely between the two loose chocks, and the screw points are entered into the pops, the tool clamps are tightened down hard upon the former, and when the screws are then advanced up to the tool it has only one degree of freedom to move. Friction between the tool and the cast-iron side-pieces, due to vertical motion of the former, is prevented by interposing greasy plates between them; and the tool is supported near its point by the strut *I* underneath it. This is of adjustable length, is formed into a knife-edge on top, and rests upon a knife-edge on the cast-steel beam *J* at bottom.

First Dynamometer, measuring Vertical Force only.

SIDE ELEVATION.

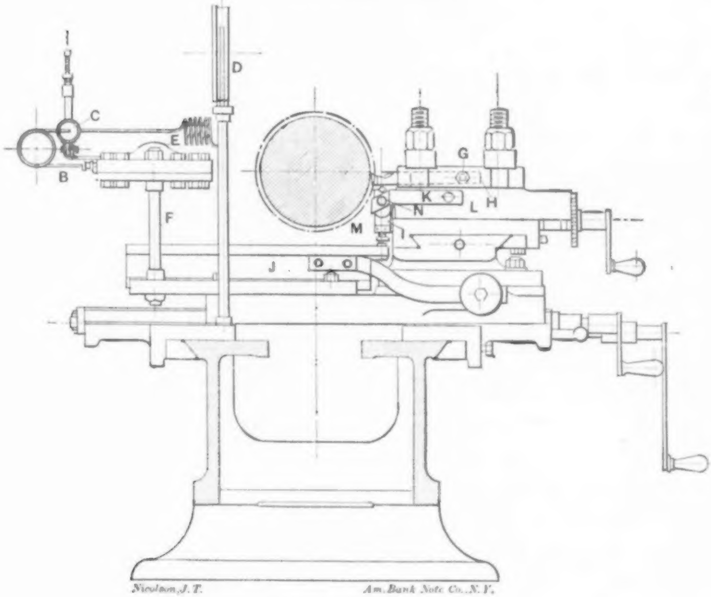


FIG. 310.

FRONT VIEW.

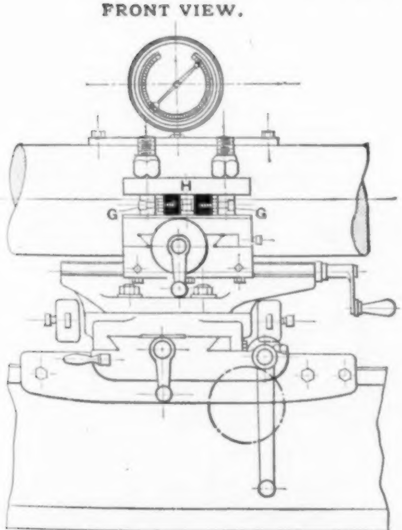


FIG. 311.

BACK VIEW.

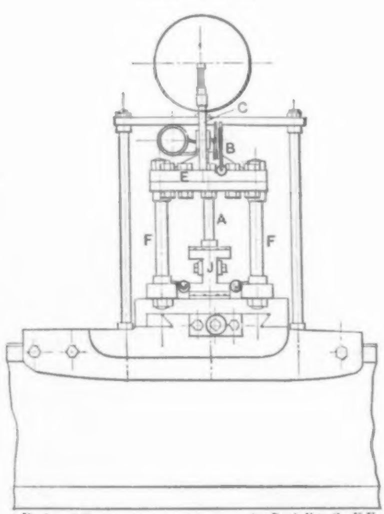


FIG. 312.

Second or Universal Dynamometer.

Sectional Elevation.

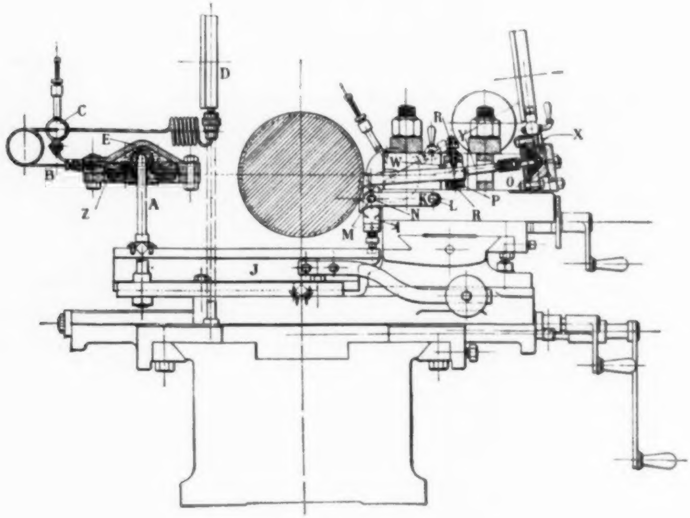


FIG. 313.

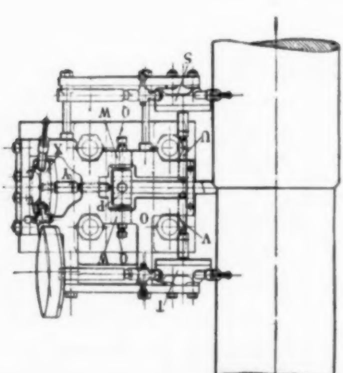


FIG. 314.—PLAN.

14. This strut I is kept in place during the cut by the stirrup-shaped piece K, which is hinged on the tap-bolts L, and retains I by means of two pointed screws M which enter the centres of the tool steel pin N connecting the double and single eye as shown.

15. The beam is about 2 feet $4\frac{1}{2}$ inches long, has a fulcrum on the under side, and another knife-edge formed upon its upper side at the opposite end for taking the diaphragm—strut A. The fulcrum is a knife-edge 4 inches long, formed on the beam and resting on a flat steel plate upon the saddle. It is important that the knife-edges should be part of the beam, so that the leverage raito may remain constantly the same, notwithstanding chatter.

16. With this arrangement loads of over 10 tons on the point of the tool have been taken when cutting, with but little more vibration than what is felt when the tool is bolted to the tool-slide in the ordinary way. The diaphragm only yields, when the fluid is air-free, by the amount necessary to supply the increased volume of the gauge due to the added pressure, and the spring of the tube is able to bring the arrangement back to zero when the load is removed. In order to assist this action the tool (whose point is set centrally to the work) is adjusted with a considerable droop; so that, as with a spring tool, the cutting force diminishes when the tool deflects.

17. In the second form of dynamometer the same large hydraulic support Z, Fig. 313, cast-steel beam J, struts A and I, are used as in the first for measuring the downward force; but three other diaphragms are added to enable the backward and side thrusts to be measured. The side elevation of the whole instrument is shown in Fig. 313, the plan in Fig. 314, and the front and side views in Figs. 315 and 316. The loose cast-iron cheek-pieces are here replaced by a casting O, Fig. 314, shaped to fit round the tool clamp studs, and recessed to provide for the gimbal tool-holder P. The object of the latter is to allow the tool freedom to move about either a horizontal axis, parallel to the lathe centre line, or a vertical axis intersecting the other. Two pointed screws Q pass through O into centre punch marks in the gimbal piece P; and two other similar screws R pass through the gimbal into pop marks on the top and bottom surfaces of the tool itself. As the tool point can now move horizontally (as well as vertically), the strut I is pointed at the bottom and rests in a recess in the cast-steel beam J, instead of upon a knife-edge formed upon it as before. The stirrup piece K is now only used to retain

Second or Universal Dynamometer.

Front View.

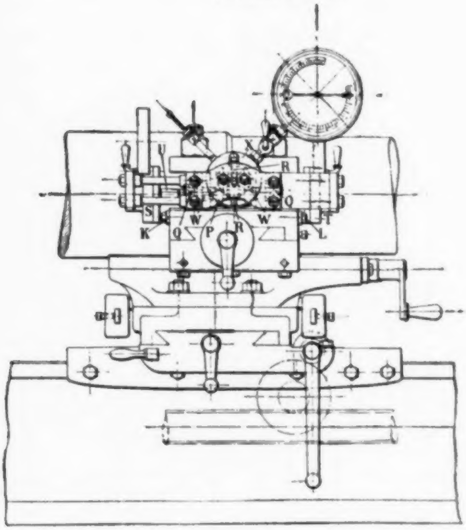


FIG. 315.

Side View.

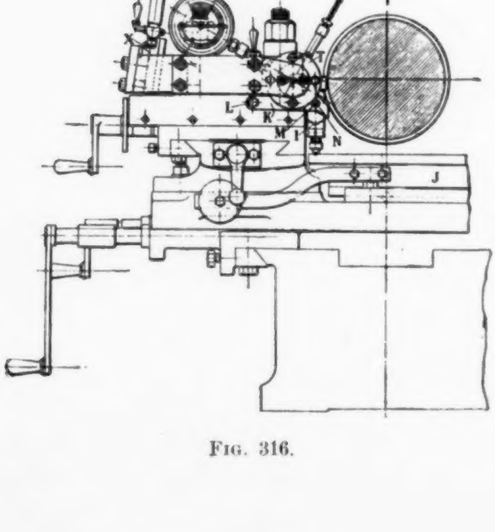


FIG. 316.

the strut I in the direction of the depth of the cut; that strut being free to tilt somewhat in the direction of the traverse feed. The two hydraulic supports S and T (with struts U and V of adjustable length) are suitably placed so as to retain the tool in a fixed position, in plan, relatively to the tool slide. When an experiment is about to be made, compression is put on both these supports S and T by screwing and lengthening struts U and V; so that the tool is firmly held sideways by two compressive forces of from 500 to 1,000 pounds each, acting upon it from the supports. If, then, the tool slide thrusts to the right when traversing, it will increase the pressure on the right-hand support T, and diminish, by the same amount, the pressure on the left-hand support S. If it should thrust to the left, *i.e.*, draw into the cut, it will have the reverse effect.

18. In order to allow of the back thrust of the tool being measured, the pointed screws Q are not tapped through the casting O itself, but through sliding blocks W fitting in slots therein. An hydraulic support X, with its strut Y, is supplied at the back of the tool; the strut centre line being inclined at the same angle as the tool, which, for the reason above mentioned, droops somewhat towards the work.

19. The tool projects very little behind the centres of screws Q, and there is thus but little vertical or horizontal motion of the point of strut Y, where it touches the tool, when the latter moves under the deflections of the diaphragms of supports Z, S or T. Before the cut commences, support X is also put under pressure by screwing and lengthening strut Y, so that the tool is kept from running in by being pressed with considerable force through the blocks W, against the front of the slats formed in the casting O. Should the tool, when taking its cut, thrust backwards with a smaller force than that corresponding to the initial pressure put upon diaphragm X, that pressure must, of course, be suitably diminished by unscrewing and shortening strut Y.

20. The general arrangement of the apparatus is shown in three photographs, Figs. 317, 318 and 319. For these the author is indebted to the kindness of his colleague, Mr. Fishenden, of the School of Technology.

Experiments Made with First Dynamometer.

21. The size of steel ordinarily used in the experiments was $1\frac{1}{4}$ inch square, although $1\frac{1}{2}$ and 2 inches square steel was some-

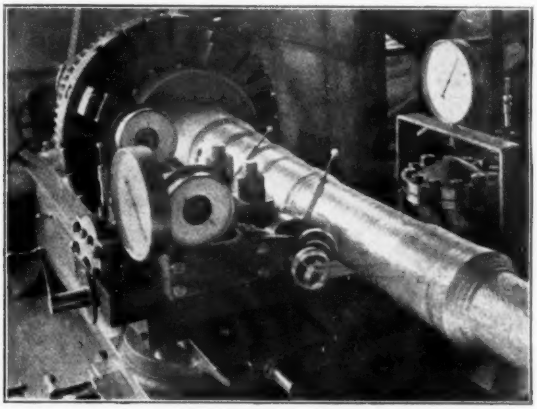


FIG. 317.
SIDE VIEW.

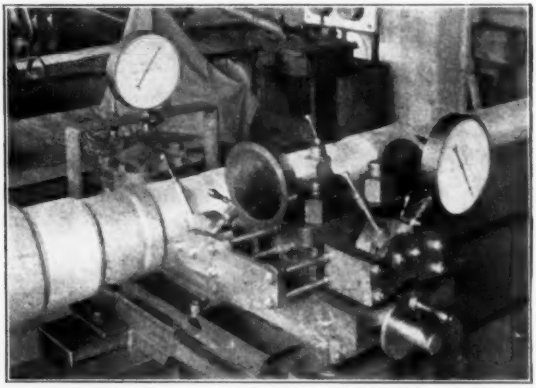


FIG. 318.
FRONT VIEW.

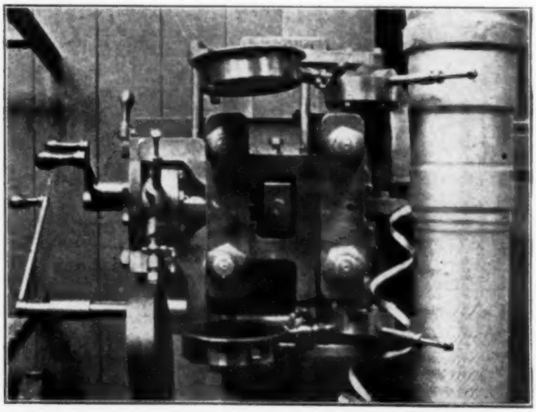


FIG. 319.
PLAN.

times employed. In projecting a series of trials upon the effect of tool angle upon cutting force, the shape of the tool point in plan required first of all to be carefully considered. Almost every variety of shape had been sent in by the steel makers in the committee's experiments, and there was no indication that any one of these was distinctly better than all the others. The round-nosed tool was the most common form, and is the easiest to forge and keep; but it has the great disadvantage from the point of view of these experiments that the actual cutting angle varies on a given tool with the depth of the cut.

22. It was therefore decided to make the cutting edge horizontal and at an angle in plan of 45 degrees with the axis of the work. The top surface of the tool was a plane containing the cutting edge, and inclined at the angle called "the cutting angle" to the vertical plane which also contained the cutting edge. The cutting edge terminated at a point $\frac{3}{8}$ inch from the right hand corner of the tool (in $1\frac{1}{4}$ inch square tools) so that the average cut taken would give a downward thrust, acting as nearly as could be arranged in the centre line of the tool, so as to prevent any twisting action and undue load upon the steel centre points. The nose of the tool had a clearance angle in plan of not less than 1 degree, and a small radius was ground on the corner between the two edges. The front clearance was 6 degrees, the tools being used with the cutting edge on the level of the centre of the work. The form of the tool end is shown in plan and elevation in Figs. 333 and 334.

23. (In the series of experiments made with the second dynamometer for elucidating the effect of different values of the angle (in plan) made by the cutting edge with the lathe centre line, the cutting edge was still kept horizontal, whether the plan angle was $22\frac{1}{2}$, 45, $67\frac{1}{2}$ or 90 degrees.)

24. The results of the series of trials made with the first dynamometer, in which the vertical component only of the cutting force was measured, are given in Tables 1, 2, 3 and 4, and have been plotted in Figs. 320 to 329.

25. Tables 1 and 2 refer to medium cast-iron; Tables 3 and 4 to soft (fluid-pressed) steel; these being the only materials so far used in these trials. The former material is somewhat harder than ordinary shop cast-iron (*vide* Manchester report), whilst the latter is a tough but not very hard steel.

TABLE I.

RESULTS OF FIRST DYNAMOMETER TRIALS ON MEDIUM CAST IRON.

(*Speed 25 Feet per Minute.*)

DATE.	No. of Experiment.	Intended.		Tool Angles.		Actual.		Area.	Observed Vertical.		Cutting Speed.	Porse-power required.	REMARKS.
		Cut.	Trav.	Plan.	Cutting.	Cut.	Trav.		Force Lbs.	Stress Tons.			
1	2	3	4	5	6	7	8	9	10	11	12	13	14
		Inch.	*Inch.*	*Deg.*	*Deg.*	*Inch.*	*Inch.*	*Square Inch.*		*Square Inch.*	*Feet per min.*		
1903. Nov. 21.....	501	1/8	1/8	45	45	0.101	0.125	0.0126	2,128	75.4	24.8	1 6	Fresh tool (failed).
"	502	1/8	1/8	45	60	0.1385	0.125	0.0173	2,186	56.6	24.7	1.680	Same tool reground.
"	503	1/4	1/8	45	60	0.2305	0.125	0.0288	3,740	58	24.55	2.78	Same tool unground.
"	504	1/8	1/8	45	75	0.1130	0.125	0.0141	2,705	85.8	24.78	2.63	Fresh tool.
"	505	1/4	1/8	45	75	0.2210	0.125	0.0276	4,775	77.3	24.58	3.55	Same tool unground.
"	506	3/8	1/8	45	75	0.3280	0.125	0.0410	6,980	76 2	24.36	5.15	" " "
"	507	1/2	1/8	45	75	0.4470	0.125	0.0559	10,230	82.5	24.13	7.47	" " "
Nov. 24.....	508	3/8	1/8	45	60	0.3620	0.125	0.0453	5,400	53.2	24.3	3.98	Fresh tool.
"	509	1/2	1/8	45	60	0.4700	0.125	0.0587	7,700	58.5	24.1	5 62	Same tool unground.
"	510	1/8	1/8	45	90	0.1000	0.125	0.0125	2,760	98.5	24.8	2.075	Fresh tool.
"	511	1/4	1/8	45	90	0.2010	0.125	0.0250	5,175	92.5	24.58	3.85	Same tool unground.
"	512	3/8	1/8	45	90	0.3070	0.125	0.0384	8,340	97.2	24 35	6.16	" " (broke).
"	513	1/4	1/8	45	45	0.2260	0.125	0.0283	4,450	70	24.53	3.305	Fresh ground.
"	514	3/8	1/8	45	45	0.3120	0.125	0.0390	6,780	77.5	24.36	5.0	Same tool unground (speeded up and failed).

Nov. 25.....	515	1/8	1/16	45	48½	0.1340	0.0625	0.0084	1,325	93.5	24.7	0.995	Fresh tool.
"	516	1/4	1/16	45	48½	0.2500	0.0625	0.0156	2,188	62.5	25.5	1.62	Same tool unground.
"	517	3/8	1/16	45	48½	0.3840	0.0625	0.0240	3,190	59.3	24.18	2.33	" " "
"	518	1/2	1/16	45	48½	0.4880	0.0625	0.0305	4,240	62.1	24	3.2	" " "
"	519	1/8	1/16	45	60	0.1160	0.0625	0.00726	1,173	72.1	24.7	0.88	Fresh tool.
"	520	1/4	1/16	45	60	0.2440	0.0625	0.01523	2,186	64.2	24.5	1.622	Same tool unground.
"	521	3/8	1/16	45	60	0.3620	0.0625	0.0226	3,106	61.4	24.25	2.29	" " "
"	522	1/2	1/16	45	60	0.5000	0.0625	0.03125	4,486	64.2	23.9	3.24	" " "
Dec. 1.....	523	1/8	1/16	45	75	0.1260	0.0625	0.00781	1,437	82.5	24.75	1.08	Fresh tool.
"	524	1/4	1/16	45	75	0.2375	0.0625	0.01484	2,474	74.4	24.4	1.865	Same tool unground.
"	525	3/8	1/16	45	75	0.3635	0.0625	0.02271	3,738	73.6	24.22	2.75	" " "
"	526	1/2	1/16	45	75	0.4975	0.0625	0.03110	5,350	77.0	23.9	3.86	Tool reground.
"	527	1/8	1/16	45	90	0.1200	0.0625	0.00750	1,840	109.5	24.8	1.38	Fresh tool.
"	528	1/4	1/16	45	90	0.2245	0.0625	0.01403	3,278	104.1	24.4	2.42	Same tool reground.
"	529	3/8	1/16	45	90	0.3430	0.0625	0.02143	4,658	97.6	24.2	3.42	" " "
"	530	1/2	1/16	45	90	0.4660	0.0625	0.02912	6,152	94.5	23.9	4.46	" " "
Dec. 8.....	531	1/8	1/4	45	75	0.0885	0.25	0.02212	3,680	74.3	24.82	2.77	Fresh tool.
"	532	1/8	1/4	45	60	0.1055	0.25	0.02638	3,473	56.6	24.8	2.58	" "
"	533	1/4	1/4	45	60	0.2300	0.25	0.05750	6,525	50.7	24.55	4.84	Same tool reground.
"	534	3/8	1/4	45	60	0.2940	0.25	0.07350	8,740	53	24.4	6.47	" " "
"	535	1/4	1/4	45	75½	0.1750	0.25	0.04375	6,875	70.3	24.65	5.13	Fresh tool.
"	536	1/8	1/4	45	90	0.1060	0.25	0.0265	4,740	80	24.8	3.57	" "
"	537	1/4	1/4	45	90	0.1835	0.25	0.0459	8,110	79	24.6	6.04	Same tool unground.
Dec. 10.....	538	1/8	1/4	45	47.5	0.1010	0.25	0.0252	3,230	57.2	24.8	2.425	Fresh tool.
"	539	1/4	1/4	45	46	0.2180	0.25	0.0545	6,570	53.6	24.5	4.88	Same tool unground.
"	540	3/8	1/4	45	75.5	0.3300	0.25	0.0825	10,350	54	24.3	7.62	Fresh tool.
Dec. 11.....	541	1/8	3/8	45	75.5	0.1200	0.375	0.045	6,060	60.3	24.75	4.55	Same tool unground.
"	542	1/8	3/8	45	61	0.0805	0.375	0.0302	5,180	77	24.8	3.893	Fresh tool.
"	543	1/8	3/8	45	47	0.0780	0.375	0.0292	5,330	81.6	24.8	4.02	" "
"	544	1/8	3/8	45	90	0.0782	0.375	0.0283	7,600	116	24.82	5.72	" "

26. Referring to tables 1 and 3, the first two columns give the date and number of the trial, the third and fourth the intended cut and traverse, whilst the fifth and sixth give the tool angles, the seventh and eighth the actual cut and traverse, and the ninth gives the product of these, called the area of the cut. Column 10 records the vertical force actually observed upon the tool point, whilst number 11 gives the cutting stress, being the quotient of the cutting force (col. 10) by the area of the cut (col. 9). Column 12 gives the actual cutting speed, and column 13 the horse-power required for cutting, being the products of the numbers in columns 10 and 12 divided by 33,000. The last column contains remarks as to the state of the tool at the commencement of each trial and other points of special interest.

27. For both cast iron and steel it was the intention to make trials with each of four different traverses: $\frac{1}{16}$ inch, $\frac{1}{8}$ inch, $\frac{1}{4}$ inch, and $\frac{3}{8}$ inch, and with four depths of cut for each traverse: $\frac{1}{8}$ inch, $\frac{1}{4}$ inch, $\frac{3}{8}$ inch, and $\frac{1}{2}$ inch. This scheme was carried out in the case of the cast-iron, so far as was possible with the means available, for each of the four cutting angles of 45, 60, 75, and 90 degrees.

28. Table 2 indicates the scope of the cast-iron series, and records the cutting stresses observed, in tons per square inch, for each series of cuts and each of the four cutting angles employed.

29. The cutting stresses tabulated in Table 2, which are obtained by dividing the observed vertical cutting forces by 2,240 and by the area of the cut, as given in Table 1 (columns 10 and 9), have been averaged for each traverse and tool angle in Figs. 320, 321, 322 and 323, and the results obtained have been plotted in Fig. 328, as ordinates on a base of cutting angles.

30. This plate indicates a somewhat lower stress for wide than for fine traverses, although this concluson does not appear to hold in its entirety, especially for the keenest cutting angle used (45 degrees). It may be pointed out that the spots plotted in Figs. 328 and 329 are not single experiments, but are the average stresses for all depths of cut deduced from the sloping straight lines of Figs. 320 to 323 (or Figs. 324 to 327 for steel). These sloping lines are drawn so as to allow for the differing degree of sharpness of the tool used in each experiment depending on the number of previous runs it had had. [The small figures beside each spot on the last-mentioned figures indicate the number of trials on which the tool had already been used without regrinding.]

31. The lines are drawn straight in Figs. 320 to 322, as expressing the conclusion, which is the simplest that can be obtained as a first approximation to the observations, that, for a given traverse, the cutting force is simply proportional to the depth of cut, or that the cutting stress is constant for a given width of traverse and given tool-angle. The positions of the spots in Fig. 328, may therefore be viewed with some degree of confidence in regard to their accuracy.

TABLE II.

Medium Cast Iron Trials.

(Speed 25 Feet per Minute.)

Numbers of Trials for Reference as given in Table I. and Cutting Stress deduced.

Traverses		1/16 inch.				1/8 inch.				1/4 inch.				3/8 inch.		
Cuts		1/8	1/4	3/8	1/2	1/8	1/4	3/8	1/2	1/8	1/4	3/8	1/2	1/8	1/4	3/8
Tool Angles (cutting)	45°	515 93.5	516 62.5	517 59.3	518 62.1	501 25.4	513 70	514 77.5	No.	538 57.2	539 52.6	No.	No.	543 31.6	No.	No.
	60°	519 72.6	520 64.2	521 61.4	522 64.2	502 56.6	503 58	508 53.2	509 58.5	532 56.6	533 50.7	534 53	No.	542 77	No.	No.
Plan Angle 45° throughout	75°	523 82.5	524 74.4	525 75.6	526 77.0	504 85.8	505 77.3	506 76.2	507 82.5	531 74.3	535 70.3	540 54	No.	541 60.3	No.	No.
	90°	527 109.5	528 104.1	529 97.6	530 94.5	510 98.5	511 92.5	512 97.2	No.	536 80	537 79	No.	No.	544 116	No.	No.

32. The variation of the cutting stress with the cutting angle is very marked. It varies by nearly one hundred per cent. of its smallest value, which takes place, in every case, for a cutting angle of about 60 degrees. As subsequently shown, however, this angle of minimum cutting force is by no means that of greatest durability. A cutting angle of 80 degrees is that indicated as being best for shop use, and the cutting stress for this angle is about 75 tons per square inch.

33. In Table 3 the results obtained in the experiments on soft (fluid-pressed) steel are recorded in the same manner as already described for cast-iron.

34. In the series of trials with soft (fluid-pressed) steel, the scheme of trials mentioned above was commenced and half completed, as shown in Table 4, but as by that time the

TABLE III.

RESULTS OF FIRST DYNAMOMETER TRIALS ON SOFT (FLUID-PRESSED) STEEL.

(Speed 50 Feet per Minute.)

DATE.	No. of Experiment.	Intended.		Tool Angles.		Actual.		Actual Area of Cut.	Observed Vertical.		Cutting Speed.	Horse-power required.	REMARKS.
		Cut.	Trav.	Plan.	Cutting.	Cut.	Trav.		Force Lbs.	Stress Tons.			
1	2	3	4	5	6	7	8	9	10	11	12	13	14
		Inch.	*Inch.*	*Deg.*	*Deg.*	*Inch.*	*Inch.*	*Square Inch.*		*Square Inch.*	*Feet per min.*		
1904. Jan. 11.....	603	$\frac{1}{8}$	$\frac{1}{16}$	45	60	0.1270	0.0625	0.0079	1,143	64.6	49.5	1.714	Fresh tool.
"	604	$\frac{1}{4}$	$\frac{1}{16}$	45	60	0.2575	0.0625	0.0161	2,400	66.4	48.9	3.558	Same tool unground.
"	605	$\frac{3}{8}$	$\frac{1}{16}$	45	60	0.3650	0.0625	0.0228	4,050	79.5	48.35	5.94	" " " (Bad shape of tool, shavings curled).
Jan. 12.....	606	$\frac{3}{8}$	$\frac{1}{16}$	45	60	0.3750	0.0625	0.0234	3,630	69.3	48.0	5.28	Same tool ground to clear cutting.
"	607	$\frac{1}{2}$	$\frac{1}{16}$	45	60	0.5020	0.0625	0.0314	6,280	89.1	47.8	9.1	Same tool unground.
"	608	$\frac{1}{8}$	$\frac{1}{16}$	45	75	0.1245	0.0625	0.0078	1,390	79.5	49.5	2.85	Reforged and ground.
"	609	$\frac{1}{4}$	$\frac{1}{16}$	45	75	0.2405	0.0625	0.0150	3,360	100	48.8	4 97	Same tool unground.
"	610	$\frac{3}{8}$	$\frac{1}{16}$	45	75	0.3425	0.0625	0.0214	5,470	112	48.5	8.04	" " "
"	611	$\frac{1}{2}$	$\frac{1}{16}$	45	75	0.4600	0.0625	0.0 87	7,630	119	48.0	11.12	" " "
"	612	$\frac{1}{8}$	$\frac{1}{16}$	45	90	0.1145	0.0625	0.0072	2,790	173	49.6	4.19	Fresh tool.
"	613	$\frac{1}{4}$	$\frac{1}{16}$	45	90	0.2200	0.0625	0.0137	5,070	165	48.9	7 52	Same tool unground.
"	614	$\frac{3}{8}$	$\frac{1}{16}$	45	90	0.3195	0 66.5	0.0199	7,150	160	48.6	10.52	" " "
"	615	$\frac{1}{2}$	$\frac{1}{16}$	45	90	0.4175	0.0625	0.0261	9,130	156.2	48.1	13.35	" " "

Jan. 13.....	616	1/8	1/8	45	60	0.1255	0.125	0.0157	2,470	70	49.5	3.705	Fresh tool.
"	617	1/4	1/8	45	60	0.2565	0.125	0.0321	5,810	80.8	48.8	8.6	Same tool unground.
"	618	3/8	1/8	45	60	0.3570	0.125	0.0446	8,880	89	48.4	13.0	" " "
"	619	1/2	1/8	45	60	0.4876	0.125	0.0609	11,900	87.2	47.9	17.3	" " "
"	620	1/8	1/8	45	75	0.1220	0.125	0.0152	3,390	99.6	49 0	5.04	Tool reground.
"	621	1/4	1/8	45	75	0.2240	0.125	0.0280	6,930	110.6	49.0	10.3	Same tool unground.
"	622	3/8	1/8	45	75	0.3235	0.125	0.0404	9,960	110.6	48.5	14.65	" " "
"	623	1/2	1/8	45	75	0.4575	0.125	0.0572	12,700	99	48	18.5	" " "
Jan. 14.....	624	1/8	1/16	45	45	0.1300	0.0625	0.0081	1,740	95.3	49.4	2.6	Fresh tool.
"	625	1/4	1/16	45	45	0.2580	0.0625	0.0161	3,540	98	48.8	5.24	Same tool unground.
"	626	3/8	1/16	45	45	0.3880	0.0625	0.0242	5,460	102	48.4	8.0	" " "
"	627	1/2	1/16	45	45	0.5135	0.0625	0.0321	6,440	89.4	47.4	9.25	Same tool reground.
"	628	1/8	1/8	45	45	0.1200	0.125	0.0150	3,280	97.4	49.4	4.92	" " unground.
"	629	1/4	1/8	45	45	0.2480	0.125	0.0310	6,560	94.5	48.7	9.7	" " "
"	630	1/2	1/8	45	45	0.4670	0.125	0.0584	12,880	98.3	47.7	18.65	" " reground.
"	631	3/8	1/8	45	45	0.3340	0.125	0.0418	9,780	109.2	48.4	14.3	" " unground.
"	632	1/8	1/8	45	90	0.0975	0.125	0.0122	3,200	117.3	49.5	4.8	Fresh tool.
"	633	1/4	1/8	45	90	0.1905	0.125	0.0238	7,670	144.2	49	11.38	Same tool unground.
"	634	3/8	1/8	45	90	0.2730	0.125	0.0343	10,130	131.7	48 5	14.9	" " "
"	635	1/2	1/8	46	90	0.4130	0.125	0.0517	14,770	127.3	47.9	21.42	" " "

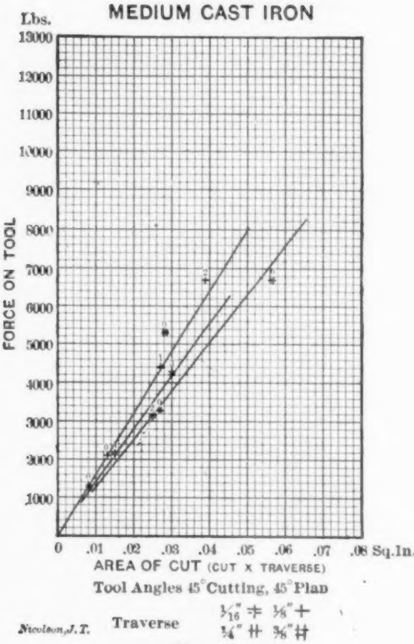


Fig. 320.

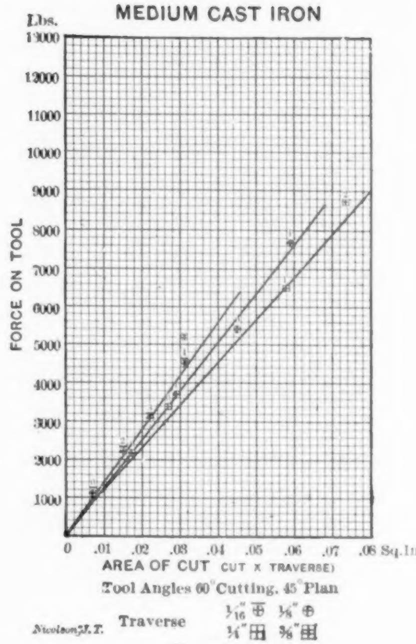


Fig. 321.

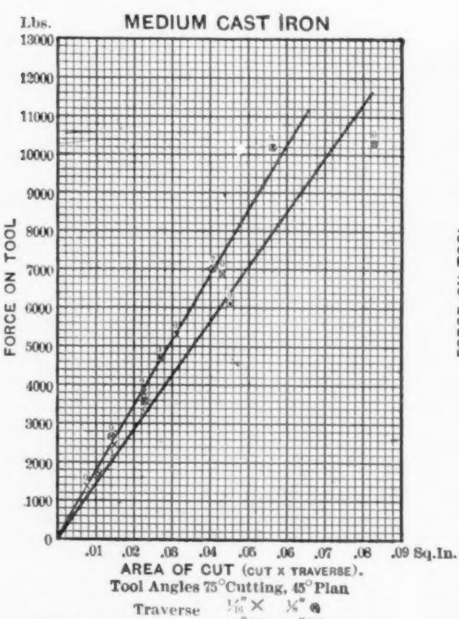


Fig. 322.

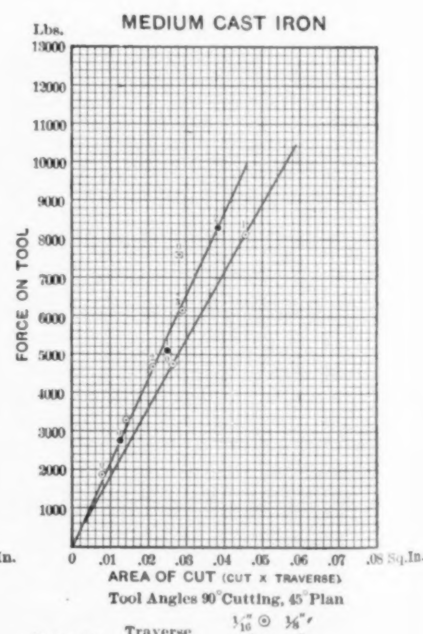


Fig. 323.

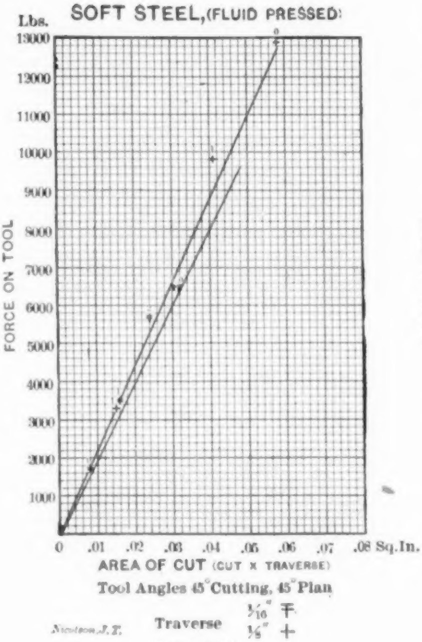


Fig. 324.

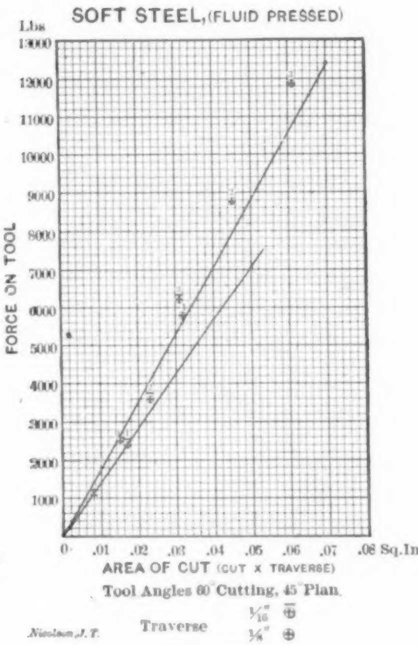


Fig. 325.

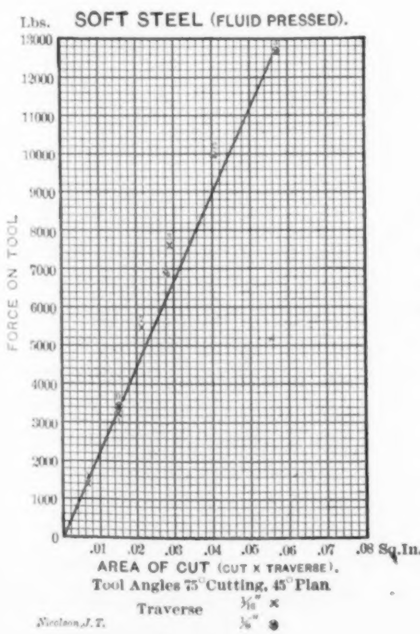


Fig. 326.

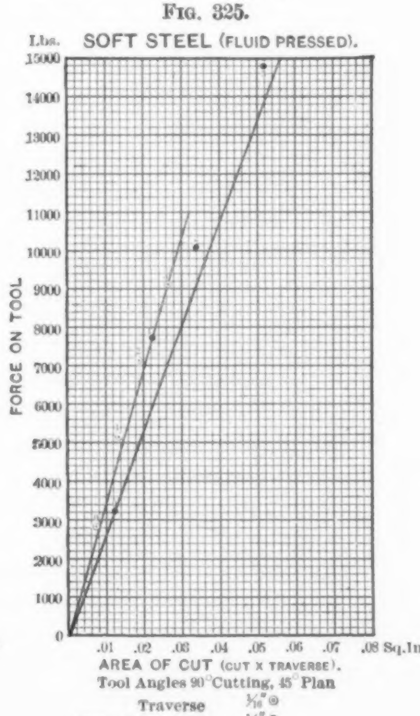


Fig. 327.

second dynamometer was ready for work a new schedule on trials was made out and executed, and will be described later on.

35. Table 4 gives for soft steel the same results as were tabulated in Table 2 for cast-iron. The cutting stresses in this table have again been averaged for each traverse and tool-angle by the method shown on Figs. 324 to 327, and the results obtained for them in this way have been plotted as ordinates on a base of tool-angles in Fig. 329.

TABLE IV.

SOFT (FLUID-PRESSED) STEEL TRIALS.

(*Speed* 50 *Feet per Minute.*)

Numbers of Trials for Reference as in Table III, and Cutting Stresses deduced.

TRAVERSES		$\frac{1}{16}$ inch.				$\frac{1}{8}$ inch.				$\frac{1}{4}$ inch.				$\frac{3}{8}$ inch.		
CUTS		$\frac{1}{8}$	$\frac{1}{4}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{1}{8}$	$\frac{1}{4}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{1}{8}$	$\frac{1}{4}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{1}{8}$	$\frac{1}{4}$	$\frac{3}{8}$
Tool Angles (cutting).	45°	624	625	626	627	629	629	631	630							
		95.3	98	102	89.4	97.4	94.5	109.2	98.3							
	60°	603	604	605	607	616	617	618	619							
		64.6	66.4	79.5	89.1	70	80.8	89	87.2							
				606												
				69.3												
Plan Angle 45° throughout.	75°	608	609	610	611	620	621	622	623							
		79.5	100	112	119	99.6	110.6	110.6	99							
	90°	612	613	614	615	632	633	634	635							
		173	165	160	156.2	117.3	144.2	131.7	127.3							

36. The variation of the cutting-stress with the traverse in the case of soft steel is somewhat complicated. For keen cutting angles (below 75 degrees) fine traverses require less cutting force than wide ones, whilst for blunt-nosed tools (*i.e.*, cutting angles greater than 75 degrees) the reverse is the case, and the fine traverse cut requires the greater effort to remove. At a cutting angle of 75 degrees the stress is the same whether the traverse be $\frac{1}{16}$ inch or $\frac{1}{8}$ inch, and has the value of about 100 tons per square inch. It is curious to remark that this angle of 75 degrees is also about the best angle for shop use, as shown by the durability trials subsequently to be cited.

37. We may, therefore, say, with some confidence, that the

ordinary shop tool, when cutting soft steel of this quality, exerts a vertical force of 100 tons per square inch of area of cut removed irrespective of the proportion of width of traverse to depth of cut. [It may be pointed out that 98.5 tons per square inch was obtained in the Manchester experiments as the average cutting stress for the endurance trials on soft steel (Table 20 of Report) in which one shape of tool was used throughout, the cutting angle being about 70 degrees. The tool is figured on page 256 of the Report. According to the results of Fig. 329, the stress on these endurance trials ought to have been about 88 tons or even less, as the speed was 90 feet instead of 50 feet per minute, but it must be remembered that the electrical method of measuring the cutting force from which the figure 98.5 was deduced includes not only the vertical work, but also that done in pushing away the shaving over the face of the tool, and ought in most cases to give a greater value for the cutting stress than that attained with the dynamometer. The agreement is therefore very close, and the two results are mutually confirmatory.]

Experiments on Durability of Different Cutting Angles.

38. All the above trials were made in the endeavor to determine the laws of the variation of *cutting force* with tool-angle and with shape of cut. It was, however, *à priori* to be expected that the tool-angle which gave the smallest cutting force would also prove the most durable or remove the greatest weight of material before failure. As this is a point of even greater practical importance than the other, two further series of trials were projected, one on the soft steel, the other with the medium cast-iron, for the purpose of finding the cutting angle to be commended for shop use.

39. In the cast-iron series a cutting speed of 44 feet per minute, with a cut $\frac{3}{16}$ inch deep by $\frac{1}{16}$ inch traverse, was decided upon, after about fifteen preliminary trials had been made. It was found in these preliminary experiments that a foot per minute, more or less, in the cutting speed made a great difference in the duration of the experiment; and, as time and material had to be economized, the careful adjustment of the speed was necessary to ensure uniform and consistent results. Cutting angles of less than 60 degrees were excluded, but it was decided to use tools of 60, 65, 70, 75, 80, 85 and 90 degrees cutting angles, and to run them at the above speed *exactly* until they failed.

The results are given in Table 5. This table contains the trial numbers and dates, the intended and actual cuts and traverses, the angles of the tools, and the time required to fail them; or the duration of the run. (The plan angle was 45 degrees throughout.)

TABLE V.

FAILURE TRIALS WITH DIFFERENT CUTTING ANGLES ON MEDIUM CAST IRON.

Intended cut, $\frac{3}{16}$ inch : traverse, $\frac{1}{16}$ inch ; cutting speed, 44 feet per minute.

DATE.	Number.	Actual		Actual Area.	Angle of Tool.	Size of Tool.	Duration of Trial.		REMARKS.
		Cut.	Traverse.						
1903.					*Deg.*	*Sq. In.*	*Min.*	*Sec.*	
Dec. 17...	561	0.1815	0.0625	0.01134	60	1¼	2	47	
" ...	562	0.1815	"	0.01134	60	1½	0	21	Too soft.
Dec. 23...	563	0.1830	"	0.01144	65	1¼	4	10	
" ...	564	0.1745	"	0.01090	65	1½	1	51	Too soft.
" ...	566	0.1875	"	0.01172	70	1¼	8	45	
" ...	567	0.1715	"	0.01071	70	1½	2	0	Too soft.
" ...	568	0.1815	"	0.01134	75	1¼	12	15	
" ...	569	0.1800	"	o.01125	75	1½	4	0	Too soft.
" ...	570	0.1770	"	0.01106	80	1¼	8	14	
1904.									
Jan. 6....	573	0.1830	"	0.01144	80	1½	5	30	Too soft.
"	574	0.1825	"	0.01141	85	1¼	7	55	
"	576	0.1710	"	0.01068	85	1½	2	40	Too soft.
"	575	0.1670	"	0.01043	90	1¼	4	50	
"	577	0.1710	"	0.01068	90	1½	1	10	Too soft.

40. The times of failure or durations of these runs are plotted as ordinates on a base of tool-angles in Fig. 330. From Table 5 and Fig. 330, it is clearly seen that a cutting angle of from 75 degrees to 80 degrees with tools of 45 degrees plan angle were the most durable for medium cast-iron. As the cut ($\frac{3}{16}$ inch) was somewhat shallow, and the tool point had a small radius (about $\frac{3}{32}$ inch) in plan, the shaving moved off in a direction nearly perpendicular to the axis of the work, instead of at right angles to the cutting edge of the tool (45 degrees in plan). This means that the actual cutting angle measured in the direction of motion of the shaving (in plan) [or the true cutting angle as per Manchester Report, p. 246] was about 81 degrees.

41. Tools should therefore be ground for maximum endurance in the cutting of cast-iron in ordinary shop practice, so that their true cutting angles are about 81 degrees, or if they are allowed 6 degrees clearance for working on the level of lathe centres, they should have an included angle of about 75 degrees.

MEDIUM CAST IRON.
VARIATION OF CUTTING STRESS WITH ANGLE OF TOOL.
(DIFFERENT TRAVERSES)

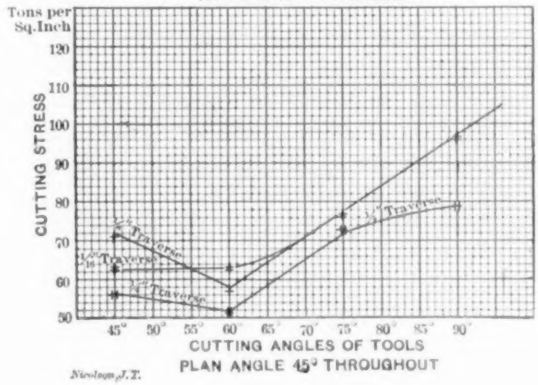


FIG. 328.

SOFT STEEL (Fluid pressed).
FAILURE TRIALS OF TOOLS WITH VARIOUS CUTTING ANGLES. CUT 1/4" TRAVERSE 1/8"
SPEED 75 FEET PER MINUTE.

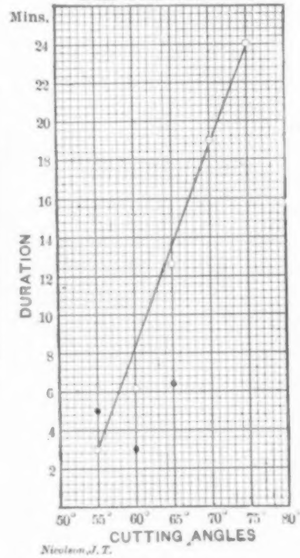


FIG. 331.

SOFT STEEL, (FLUID PRESSED).
VARIATION OF CUTTING STRESS WITH ANGLE OF TOOL.
(DIFFERENT TRAVERSES)

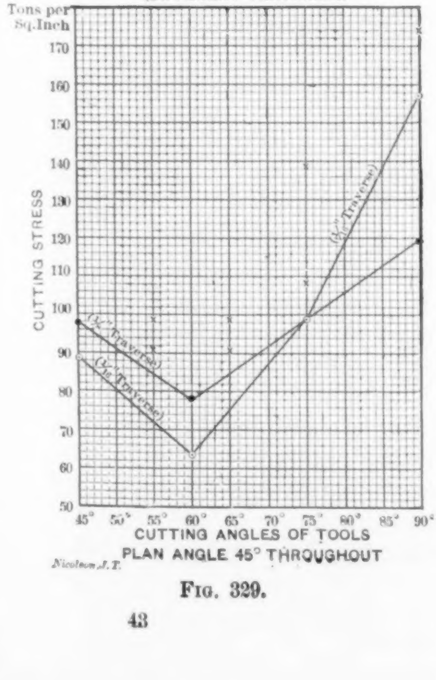


FIG. 329.

FAILURE TRIALS OF TOOLS WITH VARIOUS CUTTING ANGLES. CUT 3/16" TRAVERSE 1/16"
SPEED 44 FEET PER MINUTE.
MEDIUM CAST IRON

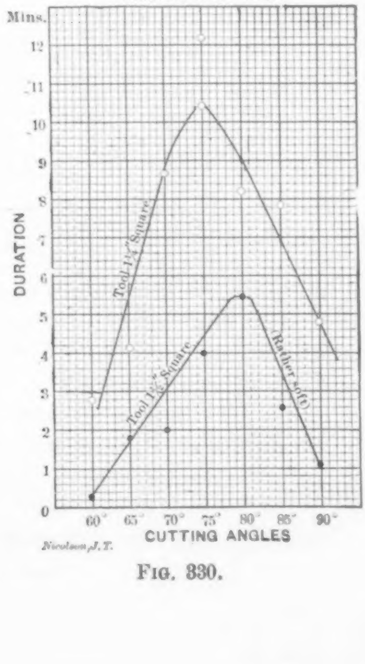


FIG. 330.

42. The series of trials made to determine the most durable angle of tool for the rapid cutting of steel had to be run at a speed of 75 feet per minute, in order to secure failure in a reasonable time on the cut of $\frac{1}{4}$ inch by $\frac{1}{8}$ inch, which had been decided upon. Unfortunately the soft-steel shaft supplied by Messrs. Whitworth (originally 22 inches diameter and 9 feet long) was, by the time the trials now referred to were commenced, reduced to a diameter of less than 6 inches in parts, and the vibration which sometimes ensued, together with the difficulty of getting sufficient length of parallel bar for a failure trial, prevented the series from giving a quite conclusive result with regard to soft steel.

Table 6 and Fig. 331 give the figures and show the nature of the results obtained.

TABLE VI.

FAILURE TRIALS WITH DIFFERENT CUTTING ANGLES ON SOFT (FLUID-PRESSED) STEEL.

Intended cut, $\frac{1}{4}$ inch; traverse, $\frac{1}{8}$ inch; cutting speed, 75 feet per minute.

DATE.	Number.	Actual Cut.	Actual Traverse.	Actual Area.	Angle of Tool.	Size of Tool.	Duration of Trial.		REMARKS.
1904.					*Deg.*	*Sq. In.*	*Min.*	*Sec.*	
March 10..	805	0.246	0.125	0.03075	55	$1\frac{1}{4}$	5	10	
" ..	806	0.2287	0.125	0.0286	60	$1\frac{1}{4}$	3	0	
" ..	807	0.229	0.125	0.02864	65	$1\frac{1}{4}$	6	15	
" ..	809	0.256	0.125	0.032	55	$1\frac{1}{4}$	3	0	
" ..	810	0.267	0.125	0.0334	60	$1\frac{1}{4}$	6	35	
" ..	811	0.2495	0.125	0.0314	65	$1\frac{1}{4}$	12	20	
" ..	812	0.2415	0.125	0.03015	70	$1\frac{1}{4}$	19	0	
" ..	813	0.2067	0.125	0.02585	75	$1\frac{1}{4}$	19	35	Not failed.
March 18..	814	0.265	0.125	0.03315	55	$1\frac{1}{2}$	3	25	
" ..	815	0.2435	0.125	0.03045	75	$1\frac{1}{2}$	24	30	Not failed.

The series is to be repeated with medium steel.

43. A further series of trials to determine the most durable cutting angle of tool for steel was carried out with the remainder of the bar of *medium* fluid-pressed steel used in the Manchester Committee's experiments.

44. The trials above reported with the *soft steel* bar proved inconclusive in their results, as the bar had become so reduced in diameter that a run of duration sufficient to fail a tool was with difficulty attainable, and excessive springing of the work and chattering of the tool took place. The medium steel bar had,

however, still a diameter of twelve or thirteen inches, and its length allowed of long runs being taken. Two series were made, one at 74 feet per minute, the other at 73 feet per minute, cutting speeds; the cut in both cases being $\frac{1}{4}$ inch deep and $\frac{1}{8}$ inch wide. Table 7 records the results obtained.

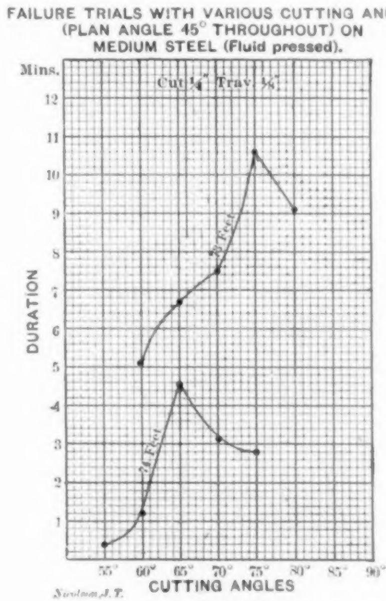


FIG. 232.

They are plotted also in Fig. 232; the durations of the various trials being set up on a base of cutting angles of the tools employed.

Taken altogether these trials seem to show that a cutting angle of about 70 degrees (included angle 65 degrees) is that which will last the longest in rapid cutting. The plan angle of the cutting edge was 45 degrees throughout.

Experiments with Universal Dynamometer.

45. The scope of the series of trials with the second or universal form of dynamometer, in which the thrusts in both horizontal directions, as well as in the vertical direction, were measured, is indicated so far as carried out up to date, in the annexed schedule.

46. One series was projected in which all the tools were to have the same cutting angle, viz., 55 degrees, but four different

plan angles of the cutting edge, viz., $22\frac{1}{2}$, 45, $67\frac{1}{2}$ and 90 degrees. In the other series a common plan angle of 45 degrees was to be preserved, with cutting angles of either 45, 60, 75 or 90 degrees.

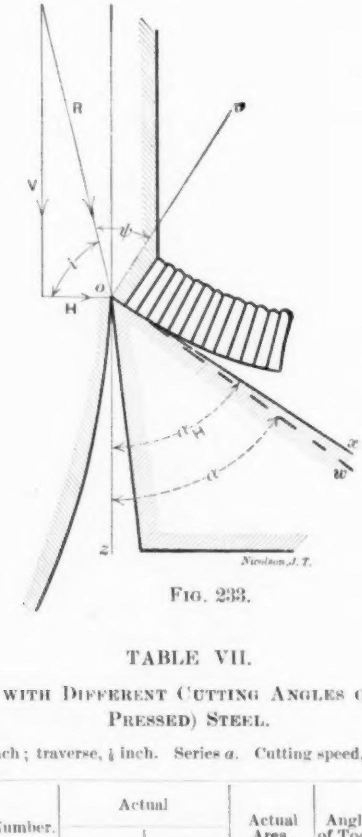


FIG. 233.

TABLE VII.

FAILURE TRIALS WITH DIFFERENT CUTTING ANGLES ON MEDIUM (FLUID-PRESSED) STEEL.

Intended cut, ¼ inch; traverse, ⅛ inch. Series *a*. Cutting speed, 74 feet per minute.

DATE.	Number.	Actual Cut.	Actual Traverse.	Actual Area.	Angle of Tool.	Duration of Trial.		REMARKS.
1904.					*Deg.*	*Min.*	*Sec.*	
March 25	818	.245	.125	.03063	55	0	25	
"	819	.2415	.125	.03018	60	1	12	
"	820	.240	.125	.03000	65	4	37	
"	821	.243	.125	.03038	70	3	10	
"	822	.233	.125	.02913	75	2	47	
Series *b*. Cutting speed, 73 feet per minute.								
March 26	823	.244	.125	.03050	60	5	10	
"	824	.240	.125	.03000	65	6	43	
"	825	.230	.125	.02875	70	7	30	
"	826	.231	.125	.02890	75	10	37	
"	827	.235	.125	.02940	80	9	8	

Experiments with Universal Dynamometers.

SCHEDULE A.

FIRST SERIES: VARIABLE PLAN ANGLE; CUTTING ANGLE, 55 DEGREES.

(*Speed, 50 Feet per Minute.*)

Number of Experiments for Reference to Tables VII and VIII. Dimensions of Cut and Traverse.

Plan Angles.													
$22\frac{1}{2}$°	746 $\frac{1}{32}\times\frac{1}{16}$	721 $\frac{1}{8}\times\frac{1}{16}$	722 $\frac{1}{4}\times\frac{1}{16}$	747*a* & *b* $\frac{3}{8}\times\frac{1}{16}$		734 $\frac{1}{8}\times\frac{1}{8}$	735 $\frac{1}{4}\times\frac{1}{8}$						
45°	717 $\frac{1}{32}\times\frac{1}{16}$	716 & 718 $\frac{1}{8}\times\frac{1}{16}$	719 $\frac{1}{4}\times\frac{1}{16}$	720 $\frac{3}{8}\times\frac{1}{16}$	711*a* $\frac{1}{64}\times\frac{1}{8}$	711*b* & *c* $\frac{1}{8}\times\frac{1}{8}$	712*a* & *b* $\frac{1}{4}\times\frac{1}{8}$	713*a* $\frac{1}{32}\times\frac{1}{8}$	713*b* $\frac{1}{16}\times\frac{1}{8}$	713*c* $\frac{1}{8}\times\frac{1}{8}$	713*d* $\frac{1}{4}\times\frac{1}{8}$	713*e* $\frac{3}{8}\times\frac{1}{8}$	713*f* $\frac{1}{2}\times\frac{1}{8}$
$67\frac{1}{2}$°	705 $\frac{1}{32}\times\frac{1}{16}$	706 $\frac{1}{8}\times\frac{1}{16}$	707*a* & *b* $\frac{1}{8}\times\frac{1}{16}$		708 $\frac{1}{4}\times\frac{1}{16}$	709 $\frac{1}{8}\times\frac{1}{8}$	710 $\frac{1}{4}\times\frac{1}{8}$	*	730 $\frac{3}{8}\times\frac{1}{8}$				
90°	714*a* & *b* $\frac{1}{16}\times\frac{1}{8}$		741 $\frac{1}{8}\times\frac{1}{8}$		715 $\frac{1}{4}\times\frac{1}{8}$		742 & 743 $\frac{3}{8}\times\frac{1}{8}$						

* A series of experiments was also made at various speeds from 1 foot in $4\frac{1}{2}$ hours up to 84 feet per minute with a tool having $67\frac{1}{2}$ degrees plan, and 55 degrees cutting angle. These tests were numbered as follows:

SCHEDULE B.

725	726	727	728	729	730	731	732	
$\frac{3}{8}\times\frac{1}{16}$	$\frac{3}{8}\times\frac{1}{8}$	Cutting $\frac{3}{8}$ inch deep, traverse $\frac{1}{8}$ inch wide.						
Dead slow.		10	20	30	45	60	84	Feet per min.

SCHEDULE C.

SECOND SERIES: VARIABLE CUTTING ANGLE; PLAN ANGLE, 45 DEGREES.

(*Speed, 50 Feet per Minute.*)

Number of Experiments for Reference to Table IX. Dimensions of Cut and Traverse.

Cutting Angles.				
45°	744 $\frac{1}{8}\times\frac{1}{8}$	745 $\frac{3}{8}\times\frac{1}{8}$		
55° or 60°	717, 716, 718, 719, 720 as above. Schedule A.			711, 712, 713 as above. Schedule A.
75°	722 *bis* $\frac{1}{8}\times\frac{1}{8}$	739 $\frac{1}{8}\times\frac{1}{8}$	740 $\frac{3}{8}\times\frac{1}{8}$	
90°	737 $\frac{1}{8}\times\frac{1}{8}$	738 $\frac{3}{16}\times\frac{1}{8}$		

47. The observations obtained in these two series of experiments (Schedules A, B and C) with the second or universal type of dynamometer give the three components of the total thrust of the work on the tool point, viz., the force acting vertically downwards (V), the force acting horizontally, parallel to the axis of the work and tending to thrust the tool to the right, or draw it in to the left (T or $-T$), and the horizontal force acting perpendicularly to the centre line of the work which tends to thrust the tool backwards (S).

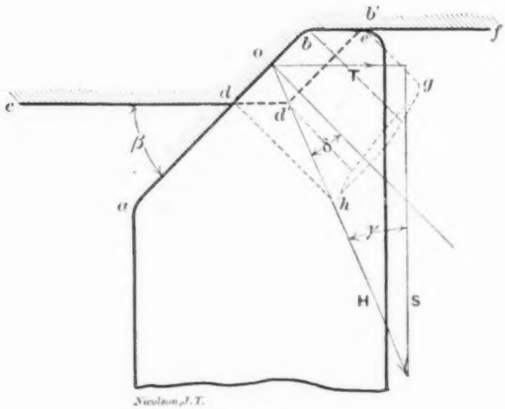


FIG. 334.

48. The knowledge of these three components enables the magnitude and direction of the resultant force (R) to be determined. Fig. 334 represents the tool and cut in plan, a b being the cutting edge, c d b f the edge of the work in section, d b e d^1 d the section of the cutting which is in course of removal. The actual point of application of R is of course unknown, but will be taken for convenience at a point o on the cutting edge midway between d and b The lines d h and b' g represent the direction of motion of the removed shaving, which, for deep cuts, was found to be perpendicular to a b.

49. From o draw T and S equal to the traversing and surfacing forces (as above described); then the resultant of these two forces (H) (acting at the angle γ to S) is the total horizontal component of resultant R.

Obviously:

$$\frac{T}{S} = \tan \gamma, \text{ and } H = S\sqrt{1 + \tan^2 \gamma}; \text{ thus } H \text{ and } \gamma \text{ are known.}$$

If we denote the plan angle of the cutting edge by β, and the angle made by H with the direction of motion of the cutting (*i.e.*, the perpendicular to ab) by δ, we have also $\delta = \beta - \gamma$.

50. Referring now to Fig. 333, which shows a vertical section of tool and work in the plane containing H (and R), the real cutting angle of the tool, called α, is shown dotted by zaw, while the angle zox, called α_H, is the cutting angle measured in the direction of the force H. Let ψ be the angle between the force R and ov perpendicular to ox. Then, drawing V and H as shown, we have:

$$\frac{V}{H} = \tan\,(180 - \overline{\alpha_H + \psi}) = \tan\,i\text{, say, where } i = 180 - (\alpha_H + \psi),$$

is the inclination of R to the horizontal; and $R = H\sqrt{1 + \tan^2 i}$.

Also $\tan \alpha_H = \dfrac{\tan \alpha}{\cos \delta}$; so that α_H and ψ are known.

51. The values of these and other quantities, which have been worked out for all the trials made on the Whitworth soft fluid-pressed steel with the universal dynamometer, are given in Tables 8, 9 and 10.

52. In these Tables, columns 1 and 2 give the number and date of the trial, columns 3 and 4 the plan and cutting angles of tool employed, columns 5 and 6 the actual depth of cut and width of traverse, column 7 the product of these, called the area of the cut, and column 8 the actual cutting speed at the mean diameter of the work. Columns 9, 10 and 11 record the results of the dynamometer observations; the traversing force (T) is the force exerted by the tool when cutting—to push the saddle and rests to the right; the surfacing force (S) is that exerted by the tool to spring the rest backwards, and the vertical force (V) is that pushing the point of the tool directly downwards. Column 12 gives the ratio of T to S from which γ, the angle of inclination to S of the resultant horizontal force (H), can be found from the formula given above. This force (H) is tabulated in column 13. From the ratio of V/H, given in column 14, the angle (i) of inclination to the horizontal of the resultant force (R), which acts upon the tool point, can be found as given above; this angle is also tabulated in column 14 and the force (R) itself in 15. Columns 16 and 17 give the cutting angle α_H of the tool measured in the direction of the force H, and the angle of inclination (ψ) of R to the normal to the top surface of the tool.

TABLE VIII.

EXPERIMENTS WITH UNIVERSAL DYNAMOMETER. VARIABLE PLAN-ANGLES.

(Soft Steel.)

Trial Number.	Date.	Tool Angle. Plan.	Tool Angle. Cutting.	Actual Depth of Cut.	Actual Width of Traverse.	Actual Area of Cut.	Cutting Speed.	Force Measurement. Traversing T.	Force Measurement. Surfacing S.	Force Measurement. Vertical V.	T/S or tan γ and γ.	Horizontal Force H.	$V/H = \tan\{180 - (a_H + \psi)\}$ and $180 - (a_H + \psi)$.	Resultant Force R.	$a_H + \psi$ and a_H.	ψ.	$T/V \times 100$.	$S/V \times 100$.	Horse-power by Dynamometer.		Cutting Stress Tons.	Remarks.
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23
	1904.	β Deg.	a Deg.	Inch.	In.	Sq. In.	Ft per Min.	Lbs.	Lbs.	Lbs.		Lbs.		Lbs.							Per Sq. In.	
746	Mar. 17	22½	55	0.033	1/16	0.0021	49.8	15	383	448	.... 0°	383	1.172 49° 30	591	130° 30 50°	75° 30	3.1	85.5	0.678	....	97	Machine ½-traverse to clean up bar and for finishing cut.
721	" 8	22½	55	0.126	1/16	0.0079	49.3	120	657	1,680	0.183 10° 30	670	2.51 68° 18	1,810	111° 42 55° 30	56° 12	7.16	39.1	2.51	....	......	0*
722	" 12	22½	55	0.276	1/16	0.0173	48.4	270	1,020	3,040	0.265 14° 50	1,060	2.87 70° 45	3,220	109° 15 55° 15	54°	8.9	34.6	4.46	....	......	2
747	" 17	22½	55	0.2385	1/16	0.0149	47.5	290	1,422	3,750	0.211 12°	1,455	2.58 68° 48	4,030	111° 12 55° 30	55° 32	7.75	38.0	5.4	....	......	2 Same as used in 746.
734	" "	22½	55	0.118	1/8	0.0148	49.25	190	1,008	3,140	0.188 10° 38	1,025	3.26 73°	3,490	109° 55° 36	53° 30	6.05	32.1	4.68	....	......	0 Fresh tool.
735	" "	22½	55	0.234	1/8	0.0293	48.2	425	1,934	6,530	0.2186 12° 20	1,990	3.285 73°	6,830	109° 55° 24	53° 36	6.5	29.6	9.5	....	......	1 Same as used in 734.
716	" 8	45	55	0.115	1/16	0.0072	49.5	78	665	1,123	....	......	......	......	......	......	7.0	58.2	......	....	......	0
718	" "	45	55	0.131	1/16	0.0082	49.5	112	711	1,344	....	......	......	......	......	......	8.35	33.0	......	....	......	2
719	" "	45	55	0.263	1/16	0.01645	48.5	320	1,026	3,250	....	......	......	......	......	......	9.9	31.7	......	....	......	3
720	" "	45	55	0.370	1/16	0.0231	48.2	510	1,215	5,150	....	......	......	......	......	......	9.93	23.7	......	....	......	4
711a	Feb. 24	45	55	0.0475	1/8	0.0059	49.6	−70	596	1,510	−0.117 −6° 38	600	2.52 69° 48	1,626	110° 12 61° 12	49°	−4.63	39.5	......	....	......	0
711b	" "	45	55	0.0655	1/8	0.0082	49.0	−51	662	1,847	−0.077 −4° 24	662	2.79 70° 18	1,965	......	......	−2.7	36	......	....	......	1
711c	" "	45	55	0.119	1/8	0.0149	49.0	+18	836	3,050	+0.022 +1° 15	836	3.64 74° 38	3,168	105° 22 63° 24	42°	+0.525	27.5	......	....	......	2
712a	" "	45	55	0.218	1/8	0.0273	48.5	240	1,560	5,710	0.154 8° 45	1,580	3.62 74° 33	5,940	105° 27 60° 36	45°	4.2	27.4	......	....	......	3
712b	" "	45	55	0.2435	1/8	0.0304	47.5	336	1,593	6,210	0.211 11° 55	1,630	3.82 75° 20	6,460	104° 40 59° 36	45°	5.4	25.7	......	....	......	4
713a	† Mar. 3	45	55	0.0325	1/8	0.0041	50	−90	423	1,008	−0.213 −12°	433	2.31 66° 36	1,095	113° 24 69° 8	44° 16	−8.9	41.8	1.53	....	......	0

† Mr. J. H. Wicksteed present. * Numbers in Remarks Column indicate the number of trials previously made by tool without regrinding.

TABLE VIII.

Experiments with Universal Dynamometer. Variable Plan-angles.

(Soft Steel.)

Trial Number.	Date.	Tool Angle. Plan.	Tool Angle. Cutting.	Actual Depth of Cut.	Actual Width of Traverse.	Actual Area of Cut.	Cutting Speed.	Force Measurements. Traversing T.	Force Measurements. Surfacing S.	Force Measurements. Vertical V.	T/S or tan γ and γ.	Horizontal Force H.	$V/H = \tan[180 - (\alpha H + \psi)]$ and $180 - (\alpha H + \psi)$.	Resulting Force R.	$\alpha H + \psi$ and αH.	ψ.	$T/V \times 100$.	$S/V \times 100$.	Horse-power by Dynamometer.		Cutting Stress Tons.	Remarks.
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23
	1904.	β Deg.	α Deg.	Inch.	In.	Sq. In.	Ft. per Min.	Lbs.	Lbs.	Lbs.		Lbs.		Lbs.							Per Sq. In	
713b †	Mar. 3	45	55	0.0645	$\frac{1}{8}$	0.0081	49.5	−84	534	1,750	−0.157 −9°	540	3.24 72°50	1,835	107°10 67°36	39°30	−4.8	30.4	2.63	....	......	1 *
713c †	" "	45	55	0.1225	$\frac{1}{8}$	0.0153	49.0	+ 6	716	3.360	0.0084 +0°30	716	4.69 78°	3,430	102° 63°40	38°20	+0.18	21.3	5.0	...	......	2
713d †	" "	45	55	0.250	$\frac{1}{8}$	0.03125	48.5	366	1,185	6,500	0.319 17°42	1,245	5.23 79°11	6,620	100° 58°10	41°50	5.64	18.3	9.55	....	......	3
713e †	" "	45	55	0.3605	$\frac{1}{8}$	0 0451	48.0	366	1,595	8,750	0.418 22°42	1,730	5.06 78°49	8,920	101°11 57°	44°11	7.62	18.3	12.70	....	......	4
713f †	" "	45	55	0.4755	$\frac{1}{8}$	0.0594	47.5	858	2,100	11,200	20.408 2°12	2,270	4.94 78°35	11,450	101°25 57°12	44°13	7.65	18.8	16.20	....	... ..	5
707	Feb. 24	67½	55	0.127	$\frac{1}{16}$	0.00795	49.5	36 2	644	1,768	0.056 3°32	644	2.74 70°	1,890	110°0 73°	37°30	2.05	36.5	2.66	...	......	0
708	" "	67½	55	0.2555	$\frac{1}{16}$	0.0160	49	211	1,118	3,670	0.169 9°36	1,140	3.22 72°45	3.850	107°15 69°45	37°30	5.75	30.5	5.45	....	......	1
709	" "	67½	55	0.118	$\frac{1}{8}$	0.0148	49.5	114	945	3,140	0.079 4°30	1,450	2.16 65°12	3,450	114°48 72°12	42°30	3.64	30.0	4.72	....	.....	2
710	" "	67½	55	0.239	$\frac{1}{8}$	0.0299	49	264	1,800	6,440	0.1372 7°48	1,940	3.32 73°14	6,720	106°46 70°42	36°	+4.1	28.0	9.57	....	.....	3
714	" "	90	55	......	$\frac{1}{8}$	....	50	150	945	1.680	....	......	......	......	......	......	......	....	2.55	....	......	6 Same tool as used for 713f.
741	Mar. 17	90	55	0.110	$\frac{1}{8}$	0.01375	49.5	178	1,602	4,340	0.111 6°18	1,602	2.64 69°18	4.540	110°42 55°0	55°42	4.1	37.0	......	....	......	0 Fresh tool.
715	Feb. 24	90	55	0.218	$\frac{1}{8}$	0.0273	49	792	2.080	6,400	0.381 20°54	2,225	2.88 70°54	6,740	109°6 76°	33°0	3.8	32.7	......	....	......	7 Same as for 714.
742	Mar. 17	90	55	Tool broke.			......	....	......	......	....	......	......	......	.....	......	......	....	......	...	......	1
743	" "	90	55	0.2885	$\frac{1}{8}$	0.0360	49	163	2,682	9.530	0.061 3°30	2,682	3.55 74°18	9,860	105°42 55°	50°42	1.7	28.0	......	...	......	0

† Mr. J. H. Wicksteed present. * Numbers in Remarks Column indicate the number of trials previously made by tool without regrinding.

TABLE IX.

EXPERIMENT WITH UNIVERSAL DYNAMOMETER. VARIABLE SPEED.

(Soft Steel.)

Trial Number.	Date.	Tool Angle.		Actual		Actual Area of Cut.	Cutting Speed.	Force Measurement.			T/S or tan γ and γ.	Horizontal Force H.	$V/H = \tan \left[180 - (\alpha_H + \psi)\right]$ and $180 - (\alpha_H + \psi)$.	Resultant Force R.	$\alpha_H + \psi$ and α_H.	ψ.	$T/V \times 100$.	$S/V \times 100$.	Horse-power by Dynamometer.		Cutting Stress Tons.	Remarks.
		Plan.	Cutting.	Depth of Cut.	Width of Traverse.			Traversing T.	Surfacing S.	Vertical V.												
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23
		β Deg.	α Deg.	Inch.	In.	Sq. In.	Ft. per Min.	Lbs.	Lbs.	Lbs.		Lbs.		Lbs.		Deg.					Per Sq. In.	
724	1904. March 14	45	45	⅜	1/16		Dead Slow Experiment.								Tool Broke.							0†
725	"	67½	55	0.3545	1/16	0.0222	Dead slow.	110	990	5,140	......	......	......	......	......	......	....	....	......	....	......	Same tool used throughout.
726	"	67½	55	0.3545	⅛	0.0443	Dead Slow.	500 to 150	1,540 to 1,360	9,060 to 8,840	......	......	......	......	......	......	....	....	......	....	113	0
727	"	67½	55	0.3545	⅛	0.0443	10	400	1,413	8,540	......	......	......	......	......	......	....	....	2.58	....	86.5	1
728	"	67½	55	0.3545	⅛	0.0443	20	405	1,539	8,540	......	......	......	......	......	......	....	....	5.16	....	86.5	2
729	"	67½	55	0.3545	⅛	0.0443	30	470	1,500	8,400	......	......	......	......	......	......	....	....	7.64	....	85.0	3
730	"	67½	55	0.3625	⅛	0.0453	45	600	1,610	8,415	0.373 20° 33	1,720	4.88 78° 30	8,630	101° 30 64° 30	37	7.13	19.2	11.45	....	83.0	4
731*a*	"	67½	55	0.3405	⅛	0.0426	60	330	1,480	8,100	......	......	......	......	......	......	....	....	14.71	?	84.9	5 Jarring badly.
731*b*	"	67½	55	0.3465	⅛	0.0433	60	330	1,250	8,780	......	......	......	......	......	......	....	....	16.00		90	6 Jarring ceased.
732	"	67½		0.3410	⅛	0.04262	84	*	1,390	7,480	......	(Tool failed in 1 min. 12 secs.)							19.0	..	78.5	7 Jarring terribly.

* No reading, as vibration closed the guage cock. † Numbers in Remarks Column indicate the number of trials previously made by tool without regrinding.

TABLE X.

EXPERIMENT WITH UNIVERSAL DYNAMOMETER. VARIABLE CUTTING ANGLES.

(Soft Steel.)

Trial Number.	Date.	Tool Angles.		Actual		Actual Area of Cut.	Cutting Speed.	Force Measurement.			T/S or tan γ and γ.	Horizontal Force H.	$V/H = \tan[180 - (\alpha H + \psi)]$ and $180 - (\alpha H + \psi)$.	Resultant Force R.	$\alpha H + \psi$ and αH.	ψ.	$T/V \times 100$.	$S/V \times 100$.	Horse-power by Dynamometer.		Cutting Stress Tons.	Remarks.
		Plan.	Cutting.	Depth of Cut.	Width of Traverse.			Traversing T.	Surfacing S.	Vertical V.												
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23
	1904.	β Deg.	α Deg.	Inch.	In.	Sq. In.	Ft. per Min.	Lbs.	Lbs.	Lbs.		Lbs.		Lbs.							Per Sq. In.	
744	March 17.	45	45	0.1255	1/8	0.0157	49	105	1,156	3,470	0.091 5°12	1,160	3.0 71°30	3,570	108°30 52°20	56°	3.0	33.4	......	......	......	0*
745	" "	45	45	0.342	1/8	0.0428	48	825	2,182	9,300	0.378 20°42	2,336	3.98 75°54	9,580	104°6 47°52	56°14	8.96	23.5	......	......	......	1
717	" "	45	55	0.030	1/16	......	......	......	......	......	......	......	......	......	......	......	......	......	......	......	......	1 as for 716.
716	" 8.	45	55	0.115	1/16	0.0072	49.5	78	665	1,123	See Table 7.	......	......	......	......	See Table 7.	7.0	58.2	......	......	......	0 fresh tool.
718	" "	45	55	0.131	1/16	0.0082	49.5	112	711	1,344		......	......	......	......		8.35	33.0	......	......	......	2 as for 717.
719	" "	45	55	0.263	1/16	0.01645	48.5	320	1,026	3,250		......	......	......	......		9.9	31.7	......	......	......	3 " 718.
720	" "	45	55	0.370	1/16	0.0231	48.2	510	1,215	5,150		......	......	......	......		9.93	23.7	......	......	......	4 " 719.
711c	Feb. 24.	45	55	0.119	1/8	0.0149	49	+18	836	3,050		......	......	......	......		0.52	27.5	......	......	......	2
712b	" "	45	55	0.2435	1/8	0.0304	48.5	336	1,593	6,210		......	......	......	......		5.4	25.7	......	......	......	4
713e	March 3.	45	55	0.3605	1/8	0.0451	48	666	1,595	8,750		......	......	......	......		7.62	18.3	12.7	......	......	4
722bis	" 14.	45	75	1/8	1/8	......	49	350	1,580	3,360	......	......	......	......	......	......	......	......	......	......	......	0
739	" 17.	45	75	0.111	1/8	0.0140	49	752	1,908	4,370	0.394 21°30	2,050	2.13 65°	4,850	115° 76°12	38°48	12.7	43.7	......	......	......	0
740	" "	45	75	0.2315	1/8	0.029	48.5	1,450	2,538	7,220	0.572 29°48	2,922	2.47 68°	7,790	112° 75°30	36°30	20	35	......	......	......	1
737	" "	45	90	0.071	1/8	0.009	50	730	2,610	3,360	0.28 15°32	2,710	1.24 51°12	4,350	128°48 90°	38°48	21.7	78	......	......	......	0
738	" "	45	90	3/16	1/8	......	......	......	3,000	......	......	......	......	......	......	......	......	......	......	......	......	1

* Numbers in Remarks Column indicate the number of trials previously made by tool without regrinding.

53. Columns 18 and 19 give items of some practical importance, viz., the ratio which T and S bear to V. As a full knowledge of V for many cuts and traverses has been given from the results of the experiments with the first dynamometer, the percentage ratios now given will enable the surfacing and traversing forces themselves for those shapes and areas of cut to be found.

54. It is not, however, to be assumed that T/V and S/V have the same values for the cutting of cast iron as those now given, which are for soft (fluid-pressed) steel; further experiments are required to determine these values for cast iron.

55. Column 20 gives the horse-power required for cutting, being the result of multiplying the actual cutting speed by the vertical force V, and dividing by 33,000. Column 22 gives the cutting stress in tons per square inch, and is got by dividing the vertical force (V, col. 11) by the area of the cut (col. 7) and by 2,240.

56. In the Tables the tests are arranged in the same way as in Schedules A, B and C, viz., Table 8 refers to experiments with variable plan angle of tool, the cutting angle being 55 degrees throughout. Table 9 refers to a special series made with one and the same tool ($67\frac{1}{2}$ degrees plan, 55 degrees cutting angle) at seven different speeds from dead slow to 84 feet per minute, the cut being $\frac{3}{8}$ inch by $\frac{1}{8}$ inch; whilst Table 10 records the results of tests made with tools of various cutting angles, the plan angle being constant throughout, and equal to 45 degrees.

57. Certain of the results in these Tables have been selected for graphic representation in Figs. 335, 336, 337, 338 and 339. Fig. 335 depicts the variation of the surfacing and traversing forces, expressed as percentages of the vertical force, with the different plan-angles of tools employed, viz., $22\frac{1}{2}$, 45, $67\frac{1}{2}$, and 90 degrees. [This angle is shown and called β in Fig. 334 and Fig. 339.] We see that the traversing force ratio varies but little and irregularly, and is of smaller importance than the surfacing force.

58. The surfacing force ratio is seen to have its smallest values for tools with a plan angle of 45 degrees (the cutting angle being 55 degrees). This minimum varies from 33 per cent. of V for light cuts to 18 per cent. of V for heavy cuts. On the other hand, the percentage sometimes rises to nearly 40 per cent.

59. In Fig. 336 curves have been drawn showing the variations of the same percentages with tools of different cutting angles, all the tools being ground with a 45 degree plan angle.

60. Both the T/V and the S/V ratios are seen to pass through minimum values in the neighborhood of 55 degrees. The minimum values of the percentage ratio of surfacing to vertical force

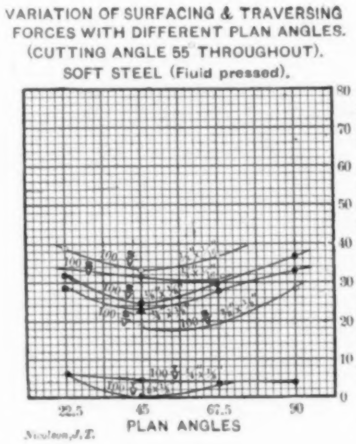


FIG. 335

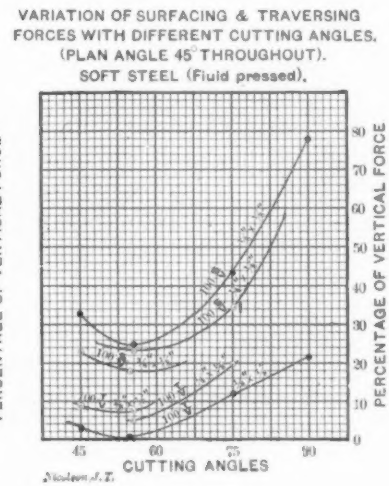


FIG. 336.

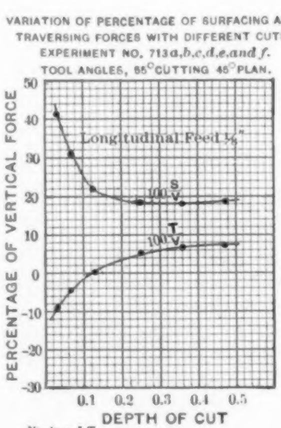


FIG. 337.

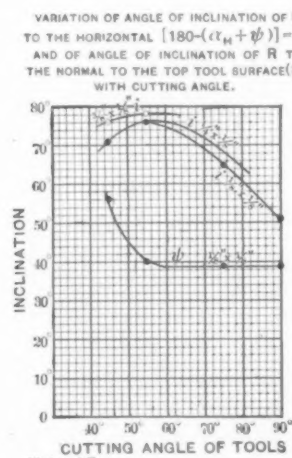


FIG. 338.

vary from 25 per cent. for light to 18 per cent. for heavier cuts. With a right angle for the cutting angle and for light cuts this ratio may attain 80 per cent. of V. T/V is again not so important as S/V, but it may reach 20 per cent. with obtuse cutting angles.

61. It is curious to observe that whilst S/V diminishes as the cut gets heavier, the reverse takes place with T/V. This is clearly shown in Fig. 337, where the results of experiments No. 713 *a*, *b*, *c*, *d*, *e*, and *f*, made with a single tool, are plotted for different depths of cut from $\frac{1}{32}$ inch by $\frac{1}{8}$ inch to $\frac{1}{2}$ inch by $\frac{1}{8}$ inch (tool 45 degrees plan, 55 degrees cutting angle).

62. Fig. 338 shows how (i), the angle of inclination of R to the horizontal, alters with tools of different cutting angles (plan angle 45 degrees). It also shows how the angle of inclination (ψ) of R to the normal face of the tool is affected by changing the value of the cutting angle.

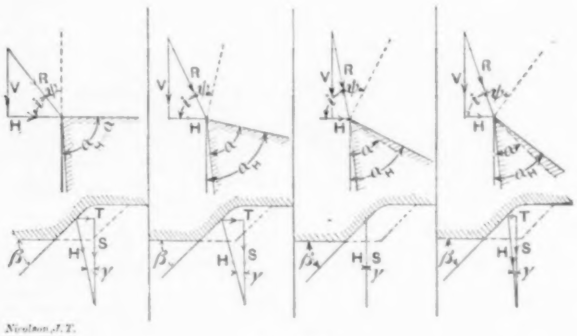


FIG. 339.

Trials 737 739 713c 744

63. The inclination of R to the perpendicular to the tool face (ψ) is remarkably constant for all tools except the keenest. It does not vary by more than one degree for tools of 55, 75 and 90 degrees cutting angle, the average value for these being about 39 degrees.

64. The angle $i = [180 - (\alpha_H + \psi)]$ at first increases and then diminishes as the cutting angle gets more acute. It attains a maximum value of 78 degrees at a cutting angle of 55 degrees, at which V, T/V and S/V are a minimum. Fig. 339 shows some of these variations in a more realistic manner. The lower and upper views are of the same kind as already described in Figs. 333 and 334.

65. The experiments (numbered 725 to 732 inclusive), the results of which are given in Table 9 are of special interest in regard to: first, the variation of the cutting force as the cut

progresses at a very low speed; second, the variation of the cutting stresses with large ranges of speed variation.

66. These experiments were made with a tool having a 55 degrees cutting and a 67½ derees plan angle; a cut $\frac{3}{8}$ inch deep by $\frac{1}{8}$ inch wide being taken.

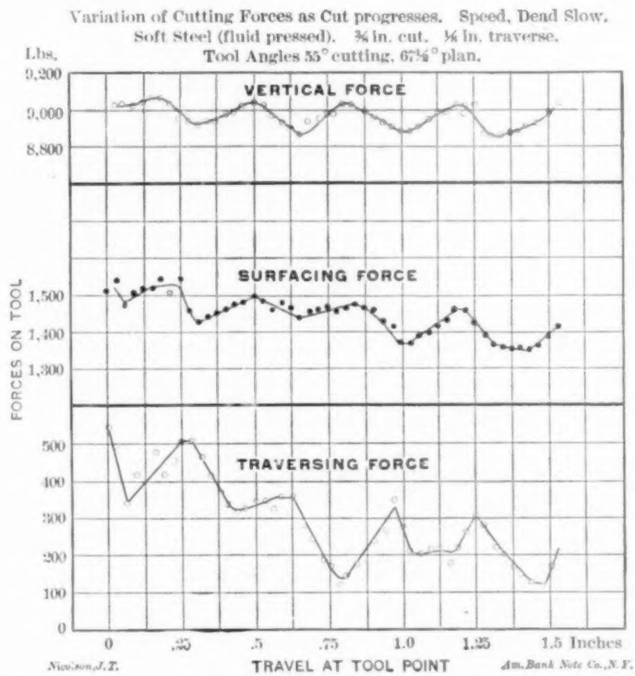


FIG. 340.

67. For numbers 725 and 726 the lathe was turned round at a cutting speed of about 1 foot in 5 hours, by means of a wire rope made fast round the large cone pulley, and hauled upon by a man operating a winch.

68. A pointer about 5 feet long was clamped upon the forging, and the four dynamometer gauges were read at every half an inch of motion of the end of this pointer, *i.e.*, at about six one-hundredths (0.0625) of an inch of the cut. The vertical force varies from 9080 to 8920 every $\frac{3}{8}$ of an inch of motion of the tool, the same wave length characterizing the variation of the surfacing and traversing forces. The observations have been plotted in Fig. 340 on a base of actual relative tool motion.

69. A similar experiment, No. 636, carried out with the first dynamometer, is shown in Fig. 341. Here the cut was heavier, $\frac{3}{8}$ inch by $\frac{1}{4}$ inch, and the tool had a 45 degree plan, and 60 degrees cutting angle. The wave length of the force-curve is about 0.6 inch for this experiment, and it varies between 13,000 and 8,000 pounds. It will be observed that the force attains a maximum

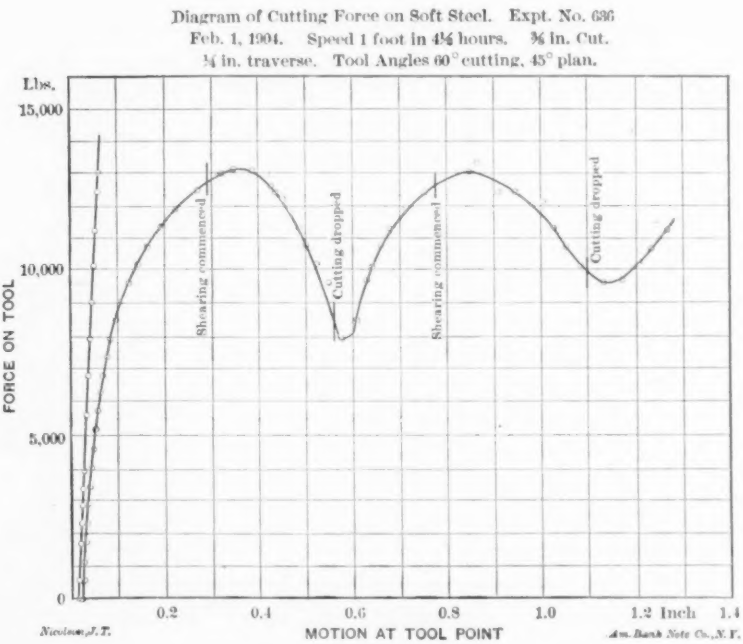


FIG. 341.

soon after the cutting commences to crack or shear across, and that it drops to a minimum when the small piece of cutting falls off the forging. At such a slow speed as this the cutting has time to shear off right across in separate fragments, whereas it forms a continuous curl of considerable rigidity when the cutting speed is higher than afew feet per minute. These fragments measured, in this experiment, about $\frac{1}{4}$ inch across the widest part of their surface next the top of the tool in the direction of motion.

70. Experiments 727 to 732 show a constantly diminishing cutting stress as the speed increases up to 84 feet per minute, notwithstanding the fact that the same tool was used upon the whole

series of cuts without regrinding. The results of these trials are shown graphically in Fig. 342. In experiment 732, when running 84 feet per minute, the tool was removing material at the rate of 12.3 pounds per minute; but failure ensued in one minute twelve seconds.

71. Further comment on the results of these experiments is reserved until the heavier cuts required to complete the series have been made.

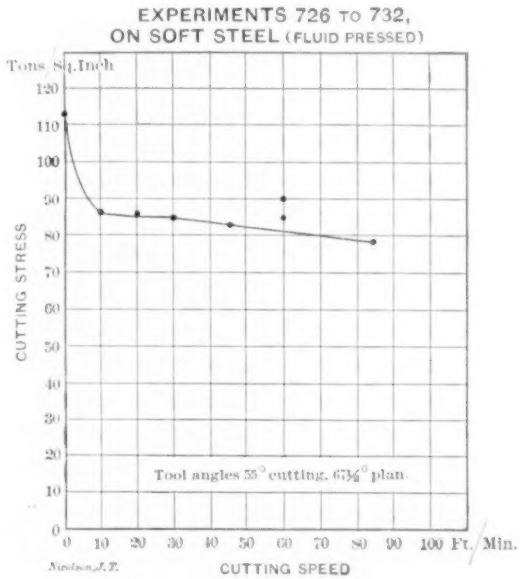


FIG. 342.

72. In conclusion, the author desires to record his great indebtedness to the following: Mr. Dempster Smith, Demonstrator of Mechanical Engineering in the School of Technology, for assistance in the laborious work of compilation and reproduction of the results. To Mr. J. T. Hodgson, Mechanical Superintendent, by whose care and ingenuity the difficulties connected with the construction and successful use of the dynamometers, which were constructed in the school workshops, were eventually overcome; and to Mr. C. Coups, turner, who carried out all the trials not only of this series, but of those for the Manchester Committee. The consistency of the results obtained is largely due to the high degree of intelligence and skill displayed by him.

DISCUSSION.

Mr. Harrington Emerson.—You have spoken of the revolutionary character of the discovery made by Messrs. White and Taylor in machine shop practice by developing these new cutting steels. In studying that question and in working with these new cutting steels I have found that they have led to something that seems to me even more valuable than their power to remove metal rapidly.

Mr. Taylor has made a very intimate time study of operations, and it was because he was making this time study that he was led to the discovery of this steel. He found that there were certain steels that cut more rapidly than others. This fact instantly fascinated Mr. Taylor, and he wanted to know why it was. His associate, Mr. White, analyzed different steels, and found there was practically no difference whatever in the analysis. The tempering treatment was next studied, until the discovery was made that when steels of certain composition are heated almost to the melting point and then cooled very rapidly they acquire the power to cut at a very high speed and at a high temperature.

But this is the point that I wish to demonstrate. Recently in an operation performed by one of these high speed steels we were able to reduce the former time of 40 hours to 20 hours; but for those 20 hours the steel was cutting only 3½ hours. Now, if we could find a steel that would cut in no time at all it would only save 3½ hours. In the other 16½ hours we could undoubtedly save, by careful analysis and study of every single operation, more than the three and a half hours that is now taken in the cutting operation. The particular value that I have found in these steels is that it leads the man who is directing operations with them to study every single part of the whole operation, the quality of the castings, the centering of the pieces, the speeds of the lathe, and so on, and if this is done it will be found that there is more time gain to be made by cutting out wastes than by the greater capacity of the steel alone. In other words, taking the same operation in two different machine shops, one machine shop with a high speed steel and the other machine shop the ordinary carbon steel, but with every single detail element toned up to its uttermost, and I venture to say that the shop with the carbon steel, but with everything else tuned up, will do faster work than the shop using high cutting steel, but go as you please in other

respects. Give, however, the well-regulated shop good steel in addition to its other economies, and the results are astonishing.

Mr. John McGeorge.—I am not surprised that our friends hesitate to discuss this paper. It requires digestion. I want to express my thanks to the author of the paper for it. It clears up points on which I have been looking for information for a long time. There is not only the value of the information on the speed of cutting, but the value of the paper consists largely in information that it gives us with regard to the requisite strength of the machine. I had to go through a little experience, not long ago, with the breaking down of one or two tools just at the time they were required for work, because they were forced a little beyond their usual capacity, and it seems to me that it was just for the want of such information as this that those tools broke down; they had not the requisite strength in the parts where it was required.

Usually manufacturers are too busy getting out orders to go into these matters, and all that I have gotten up now for is to ask if Prof. Benjamin would try and secure his share of that money that has just gone to our own institute (the Case School), to carry out fully some experiments in this direction, not only with lathes, but with regard to other tools as well, the planer, giving the side thrust and the up thrust, and in all directions; the same with the miller, and the same with the drilling machine, to find out where the stress comes exactly, because we know the directions of strains are very complicated, and we do not know which parts are going to fail first. In allowing for speeding up and the new strains that are coming on the tools made of the new steels we hardly know just where to put the extra strength, and I would draw the Professor's attention to the fact that these experiments should be made, and it would be a very useful thing for a school like his, which has contributed so much in the way of information to the profession, to carry out further this idea.

Prof. C. H. Benjamin.—I had not expected to participate in this discussion, because after reading the paper I felt that I knew so little about the subject as to make it hardly worth while to say anything. I consider this the most valuable series of experiments on the cutting power of steels that have ever been made in the history of engineering. The completeness with which they were carried out, the universal character of the dynamometer and the resolution of the cutting pressure into its ele-

ments—longitudinal, radial and tangential—are all worthy of mention.

What Mr. George has just said in reference to making experiments of this kind in this country at some place where there is time to devote to it, has led me to say a few words on that subject. It had been my intention to carry on some experiments the coming year on the high-speed steels in the laboratory of the school, and, for that purpose, to buy some special machinery. I feel, however, that those experiments would be entirely incomplete without some such apparatus as has been described in the paper which has just been read, and that to make such experiments with the ordinary forms of dynamometer that have been used in the past would be puerile. I do not know what the possibilities will be in the way of procuring or of building a dynamometer of this character which can be used on lathes, on planers, on boring mills and the other types of machine. I hope, however, that means will be provided in some way to accomplish this, because there is no question that the use of high-speed steels is one of the recent factors in our industrial development which has done more to increase the output of the machine shop than any other one element. If there are any manufacturers here who are interested in this question and in the further solution of the problem I would be glad to get in touch with them, because it is only by collaboration and co-operation that we can hope to accomplish anything. I wish some public-spirited citizen would import one of these dynamometers from Great Britain and give us the benefit of it.

Mr. Wm. Pilton.—A recent editorial comment very truly judges these experiments as "far and away the most complete that have been undertaken in any country." These experiments are of special value in that Professor Nicolson has taken practical cuts both as to depth and the feed with effective cutting speeds of about fifty feet per minute, it being clearly shown in his endurance or "failure trials" that the tool failed very quickly at seventy-five feet per minute with heavy roughing cuts.

Quoting from the experience of one who has tried about every make of high-speed steel, "We have cut a $\frac{1}{16}$ chip $\frac{1}{32}$ feed at a rate of 260 per minute from a rod of steel. Such speeds are possible for short periods, but whoever buys a rapid cutting steel with the expectation of maintaining such speed will be sadly disappointed." He further states, "We find that on steel

where there is no considerable thickness of metal to remove, a speed of one hundred feet is very satisfactory."

I know, from the actual tests and the lathe-room practice as obtained by the Niles Tool Works Co. of Hamilton, Ohio, that we get such heavy cuts and feeds at from 35 to 50 feet per minute using the modern tool steels, turning high carbon steel, forged shafts, pinions, etc. The tool point is almost a red heat and the chips beautifully colored.

The lathes we are doing this work on are the same machines as formerly used with the tools of carbon steels and slower cutting speeds. There has not been one special lathe installed in this turning department, but the speed of the line shafts has been increased, and also the countershafts, so as to get correct turning speed on suitable diameters of work and always retain the power of the back gearing. For instance: An ordinary 24-inch lathe, with its countershaft increased in speed to suit, will take very heavy cuts and feeds on work of from 3 inches to 9 inches diameter with the back gears in, and by using a second countershaft speed the minimum diameter can be further reduced.

At a meeting of the St. Louis Engineer's Club, January 7, 1903, discussing a paper by Mr. W. A. Layman on speed control; Mr. W. Cooper said in considering the fact that many tools are not designed to stand the high speeds that modern tool steel will permit: "There is no reason why a machine tool that is adapted to do a certain work should not do this work at two or three times the speed. The reason for this seems obvious, in the fact that the strains on a machine are due entirely to the torque required to make a given cut. With this given cut the speed may be increased three or four times without producing any greater strains on the machine itself, because the torque remains constant."

Of course we are building very heavy and powerfully-geared boring mills, wheel lathes, etc., especially intended for use with high-speed steel and to carry heavy cuts. In these machines the cast-iron gears are replaced with those of steel casting and steel forged gears. The belt width and the gearing ratio are also increased.

Power is required where heavy cuts and feeds on steel castings are specified on orders for certain machine tools. A nominal power at the tool of 5,000 or 6,000 pounds for each $\frac{1}{32}$ inch

of chip area is required for turning soft steel castings. This increases with the harder grades.

As to cast iron. It is more difficult to arrive at a nominal power, this because of the varying degrees of hardness, in some instances the horse-power per cubic inch of metal removed being almost equal to that of some steel castings.

In further discussing this paper, I might offer some data giving the results of several carefully planned modern machine tool tests where the results obtained compare very favorably with the results here given, but such data would not, I think, add much, if any, to the value of the paper. Professor Nicolson has stated that we may expect to hear from him when "the heavier cuts required to complete the series have been made."

*Mr. J. Hartley Wickstead.**—I think it states in the beginning of the paper that the whole of this new movement for rapid cutting was started by Messrs. Taylor and White, of the Bethlehem Steel Company, in the United States. They made a most remarkable exhibition in Paris, at the Exposition, in which the engineering world was astonished to see, instead of the deliberate manner in which steel was formerly removed by tooling in a lathe, the cuttings coming off in long continuous pieces colored purple with heat and to be able to see that, underneath the cutting, the tool was glowing with a dull red. It was about as surprising as when it was discovered that a carbon filament could be used for electric current without consuming itself. The whole idea of a tool preserving its edge when in a state of dull red heat was, I think, to all of us a revelation, and it had a most stimulating effect upon these gentlemen and others, because as soon as people knew that it was possible to do it a number of persons found they were able to do what before that had been thought impossible. Now I am particularly engaged in tool making, and I consider that it is by far the greatest revolution that has taken place in my life. When the thing came out it was not understood as to how steel could be made to do such work, nor was it understood what conditions would have to be used to take advantage of it. It was not known, for instance, whether according to the law of the flow of solids you could remove the material with a smaller pressure at a slow speed than you could at a quick speed. That is settled crucially by Dr. Nicolson, who

* Member of the Institution of Mechanical Engineers of England.

made one experiment in which he only moved the work round at the rate of one foot in four and one-half hours. It did not come off with any less pressure on the point of the tool than it did when he was moving it at the rate of 50 feet per minute. Incidentally in that experiment he found out, what is very important to know, the cause of the vibrations that are set up in removing the shaving: he was able to take a diagram of the pressure of the tool at the different stages, and he found that first of all the tool compressed a flake of metal, that is, set up a plane of cleavage, and then shot it off like the scale of a fish, and that the diagram was a diagram of very tall peaks, 13 above zero of pressure and valley only 8 above zero, the pressure on the tool rising to a certain point, and then going down, all of which you will find in the paper. That shows that you have to provide for vibrations, and that is the explanation why it is that you cannot make good use of the new steel by simply speeding up existing lathes, at least, not in all cases, for this steel not only gives you the power of going at a greater speed, but it is a steel which will preserve its edge better and will enable you to take deeper cuts and greater travel. A paper was read at the Institution of Mechanical Engineers, in London, upon the cutting angles of tools, in which a tool dynamometer had been used, and this tool dynamometer showed certain results which did not conform to the results shown by other experimenters who had arrived at their results by measuring the power that went into the lathe head-stock. Dr. Nicolson, however, has devised a very improved tool dynamometer which gives its results almost as accurately as a testing machine with which you test the strength of materials. It measures the force downward and the backward thrust of the tool. Then, together with this tool dynamometer, he had a belt dynamometer that measured the power that was going through the belt, and also the ammeter, and he was able to put the whole of these results together and there were no discrepancies; there were differences, but they were accounted for, but the net value on the tool was all confirmed by the records of the ammeter and by the belt dynamometer. Dr. Nicolson designed his dynamometer to give him a load up to 15 tons pressure on the tool, with which you can take presumably a pretty fair cut. He carried out his experiments, I think, to about half that; but he has not finished yet. He will proceed to carry the experiments further, and with still deeper cuts. I think what he has arrived at so far is that he has removed

successfully 9½ pounds weight per minute, and has found a cutting stress of 90 tons per square inch section of the cut upon soft fluid compressed steel. I think he has used about 19½ gross horse-power in doing that, and he has found a very large difference in the horse-power that is absorbed resulting from the different angles to which the tools are ground.

So that incidentally this paper is extremely valuable in giving the best angles for tools, both in regard to endurance and also in regard to the power that is absorbed.

There is one other striking remark that I find in this paper, and which seems to point to a critical speed. It is found on page 657 as follows: "It was found in these preliminary experiments that one foot per minute, more or less, in cutting speed made a great difference in the duration of the experiment." That is to say, if your tool does not last as long as you would wish it to, it does not follow that it is because you are cutting too fast; it may be because you are cutting too slow. I am not sure what the material was as to which that remark is made, but it does seem to me a point for very interesting investigation to find out more about it.

*Mr. Chambers.**—That was cast iron material.

President Wicksteed.—Thank you. Perhaps that accounts for a great deal of variation in people's opinions as to the speed at which you can cut cast iron.

I will now ask Mr. Adamson to reply.

*Mr. D. Adamson.**—I will only say that I thank you very much for the discussion, and that Dr. Nicolson and the Committee who were associated with him in the earlier experiments—referred to in the article in the *American Machinist* mentioned by the last speaker, and to which this paper is a sort of supplement—all feel that we are only at the beginning of this interesting investigation, and we shall be very glad to hear of any further experiments being carried out, either here or elsewhere. There is ample room for a very instructive series of experiments.

Prof. Wm. T. Magruder.—A somewhat similar set of experiments to those here recorded were made in 1900 under my direction by Mr. W. A. Knight, at that time instructor and at present Assistant Professor of Machine Shop Practice, at the Ohio State University, as a thesis for the degree of Mechanical Engineer,

---

* **Member of the Institution of Mechanical Engineers of England.**

The object sought was not solely to measure the work performed in cutting metal, but rather to measure the "workability of cast iron" and other metals.

For this purpose, a Pratt & Whitney planer was used. A specially constructed diaphragm pressure gauge was inserted in the tool-box, so as to receive the pressure exerted upon the tool by the piece of metal being cut. A cutting tool of constant shape was mounted in a tool holder of such shape that the point of the cutting tool was vertically in line below its point of suspension. The pressure received by the diaphragm of the pressure gauge was transmitted by oil to a double tube Bourdon pressure gauge and from it by a parallel motion to an inking pen which recorded the pressure required to make the cut. The record was made on a strip of paper positively moved by the planer-table.

The usual difficulties incident to such work were met and overcome, and the results were of much interest, as they showed the forces required to remove metal by a planer when run at a known speed, gave very different results with different kinds of cast iron, which differed both in character and amount from the diagrams obtained by cutting bar-brass, wrought iron, and the different steels. The uniformity of the texture of brass, the hard spots in cast iron, the seaminess of wrought iron, and even the greater homogeneity of tool steel, as compared with machinery steel, were noted by the apparatus and recorded autographically with much uniformity of result.

As the thesis giving the records is at present among the exhibits of the Agricultural and Mechanical Colleges at the Louisiana Purchase Exposition at St. Louis, I am unable to give greater details at this time.

*Dr. J. T. Nicolson.**—I wish to begin by thanking Mr. J. Hartley Wicksteed and the other speakers for the appreciatory remarks they have made regarding the work done and reported in the paper. I agree with all that has fallen from Mr. Wicksteed, with the exception of his last sentence or two. I hardly think the quotation he has made from the paper will bear the interpretation that, "if the tool does not last as long as you wish it to, . . . it may be because you are cutting too slow." It has never been found in these experiments that a tool could be made to last longer by a small increase of speed. Perhaps Mr. Wicksteed

* Author's Closure, under the Rules.

had in mind a possible synchronization of tool and work vibrations which might be destroyed by a slight change of cutting speed. This might happen; but it has not occurred in my experience.

With regard to Mr. John McGeorge and Professor Benjamin's remarks as to further experiments being made, I shall be very glad to furnish blue prints and full instructions for the construction of a dynamometer to Professor Benjamin; and would recommend that experiments be undertaken on the principal alloys used in engineering, such as gun metal, brass, phosphor bronze and the like. I do not see that the action in planing should be much different from that of tooling in a lathe. Drilling is a subject requiring study; and I have already instituted experiments in this direction.

Mr. Harrington Emerson's remarks regarding the small saving which will be effected by the use of high-speed steel, unless every detail of working is looked into, seem to me very much to the point. If three-fourths of the working time of a lathe is wasted by bad management it matters very little whether the cut be taken at 50 feet or 15 feet.

It is interesting to learn that Prof. W. T. Magruder used a "diaphragm pressure gauge" on the tool box of a planer in 1900. If the question be one of priority, it may be mentioned that the author used the method in 1894 at McGill University, and presented a diaphragm dynamometer to Cambridge University Engineering laboratory in that year.

Mr. Wm. Pilton thinks that all that is required in the lathes as now used to make them available for high-speed steel is to speed them up. That depends on how the gearing is arranged and what the present speed of the cone belting is. It seems clear that most cone belts run too slow and that speeds of 4,000 to 5,000 feet per minute in them are admissible, provided the cone is off the main spindle.

In conclusion the author tenders his best thanks to Mr. Daniel Adamson for looking after the paper upon its presentation in Chicago.

No. 1036.*

THE POWER PLANT OF THE TALL OFFICE BUILDING.

BY JAMES HOLLIS WELLS, NEW YORK, N. Y.

(Member of the Society.)

1. The designing of modern buildings has very materially changed the organization of an architect's office, and unless the expert is called in, there is associated with the architect an engineer whose specialty is the designing of the engineering features.

3. The engineer works in conjunction with the architect, and works up the scheme generally in its entirety, by no means a small part of the work.

3. The engineer works in conjunction with the architect, advises him as to all points of engineering design, and eventually, when the scheme is matured, takes over all details of construction.

4. Such an office is divided into two working parts; one of which is purely architectural and the other engineering. The engineering force again is divided into construction, mechanical, electrical and sanitary departments, and each part and department is dependable one upon the other, so that when all designs are completed, the general contractor is able to carry out his work rapidly and intelligently.

5. It is the intention of this paper to deal only with the engineering problems which enter into the construction of a tall building, and it will be impossible to do more than deal generally even with these. Few appreciate the vast amount of work and responsibility involved, and any one of the subdivisions of engineering alone would require a volume if written about as it should be.

Construction.

6. The designing of the foundations, the steel and iron work, and of the enclosing brick and stone work, in so far as strength

* Presented at the Chicago meeting, May and June, 1904, of the American Society of Mechanical Engineers, and forming part of Volume XXV. of the *Transactions.*

FIG. 343.—SITE OF BROAD EXCHANGE BUILDING ON AUG. 11, 1900, JUST BEFORE STEEL WORK WAS COMMENCED.

This building, containing 10,100 tons of structural steel work, over 7,000,000 brick and amounting to over 7,000,000 cu. ft. in contents, was ready for tenants prior to May 1, 1901.

FIG. 344.—BROAD EXCHANGE BUILDING, NEW YORK CITY.

is concerned, is the province of the engineer of construction, whose position is like that of the bridge engineer. The problems to be solved are sometimes very hard and require large experience to handle.

7. As an example, the foundations of a portion of the buildings of The Mutual Life Insurance Company of New York, in New York City, were carried down to bed rock by means of caissons, and the footings in some cases are over 100 feet below the curb. The walls of adjoining buildings were carried to rock, and the entire site was enclosed by a concrete wall eight feet in thickness. The floor of the boiler and pump-room is thirty feet below mean high tide level and fifty feet below the curb on Cedar Street. The seapage, which has to be pumped out of a cesspool located in the boiler-room, amounts to less than one thousand gallons a day.

8. A building known as the Wall Street Exchange was recently erected in New York City. This building is about one hundred feet square, twenty-five stories or three hundred and twenty feet in height above the curb, and two stories or twenty-three feet in depth below curb level. The foundations for this building are somewhat similar to those of The Mutual Life Building. The grillage footings were set on the foundations in December, 1902, the erection of the superstructure was commenced in January and completed from curb level to roof in seven weeks. The brick walls were started in the cellar in January, and completed on March 19, 1903. The building was ready for tenants and partly occupied on May 1, 1903.

9. These are simply instances of the problem that confronts the engineer, and it is his duty to lay out the work from a broad standpoint so as to obtain the best general results.

10. There are may factors entering into the design. Wind pressure affects column and girder sections; floor loads are variable, depending on the use the building is to be put to, and throughout the entire design, economy and rapidity of construction must be considered.

Elevators.

11. Hitherto, electric elevators have been but little used in tall buildings over fifteen stories in height on account of the design not being suitable to high speed. When the speed ex-

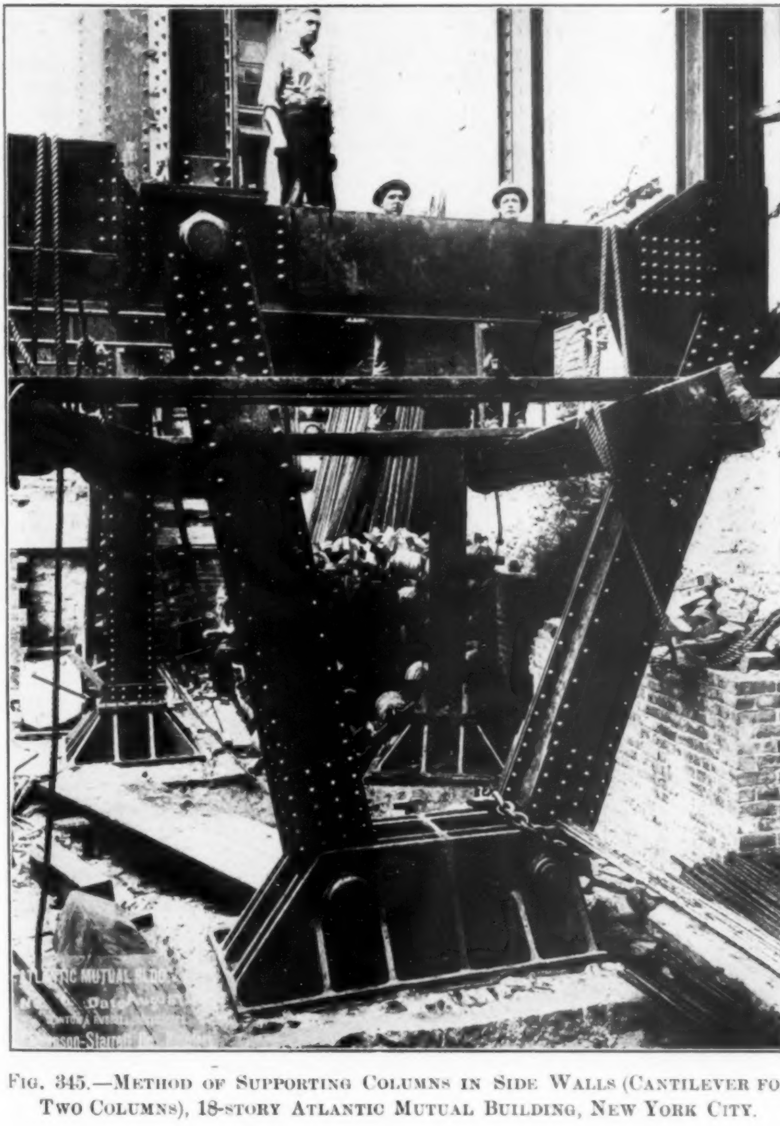


FIG. 345.—METHOD OF SUPPORTING COLUMNS IN SIDE WALLS (CANTILEVER FOR TWO COLUMNS), 18-STORY ATLANTIC MUTUAL BUILDING, NEW YORK CITY.

ceeds three hundred and fifty feet per minute, as it necessarily must in the higher buildings, hydraulic elevators running as fast as six hundred feet per minute are preferable. Modern plants are installed with either triple expansion or fly wheel pumps, and one or more compound pumps in reserve. There are, however, no absolute rules that govern, as each case must be handled independently, the conditions carefully studied and experience is the chief guide. Tests are made, of course, before the building is finally accepted from the contractors, but these tests seldom tell anything final beyond that the plant is as designed and that it will do the work economically.

12. Each elevator is usually designed to carry safely 2,500 pounds exclusive of the weight of the car, and one car is arranged to carry safes usually weighing not more than four tons at a slow speed. This car should have specially designed pawling devices to hold the car in place at any floor. Vertical cylinders for hydraulic elevators are usually preferable to horizontal cylinders on account of having shorter cables. Cold rolled steel guides are better than planed, and in fact any device that gives smooth service is desirable.

13. It is a safe general proposition when a well designed mechanical plant is installed to say that it costs less to operate an electric elevator than it does an hydraulic machine. Compound pumps generally require seventy pounds of steam per water horse-power hour, triple expansion pumps about thirty-six pounds and fly wheel pumps slightly less.

14. High-speed, single-cylinder engines are guaranteed at thirty-six pounds of steam per horse-power hour, compound engine twenty-six pounds, and steam-jacketed, four-valve, compound engines twenty pounds. An average of six and one-half water horse-power per car mile per hour is required for hydraulic elevators, and for electric elevators an average of three and one-half kilowatt hours.

15. From this data it is easy to calculate the coal and water consumption, but of course there are many other items to take into account, and experience alone teaches the trained engineer just what these items are .

16. In many installations economical operation would be considerably increased by the use of a storage battery, in which case the engine load can be kept constantly at its full load rating, which is its most economical producing point. Under these con-

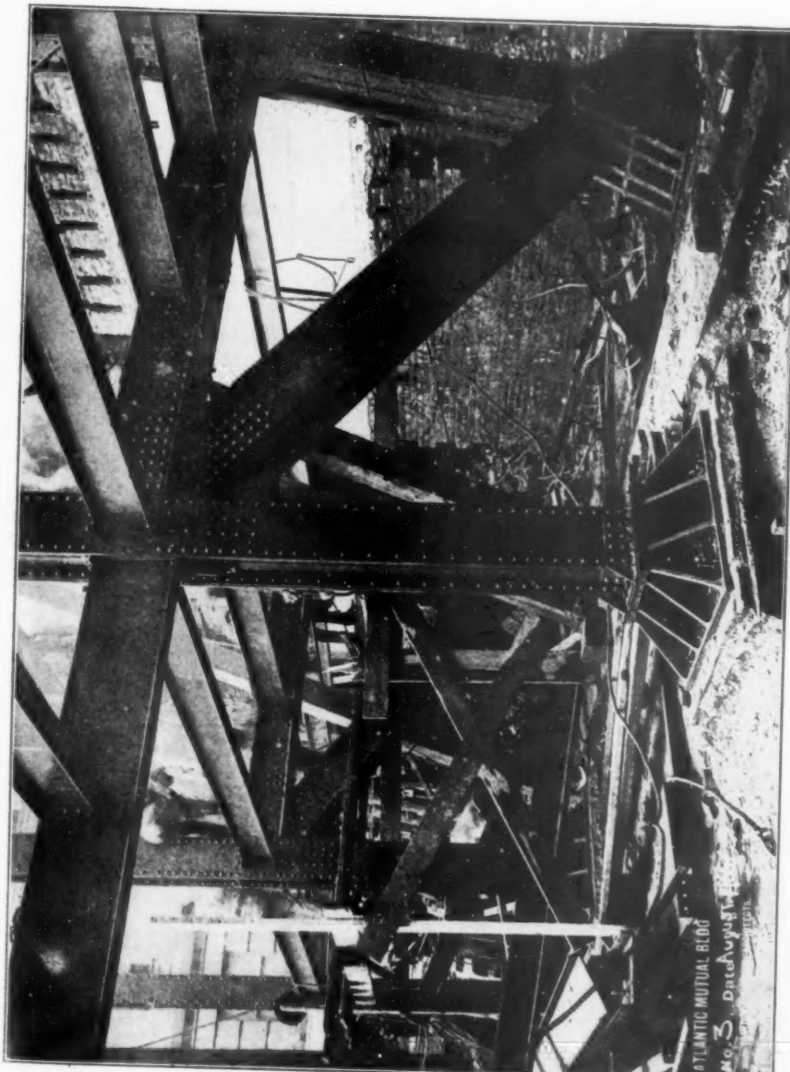


FIG. 346.—CANTILEVER SUPPORT FOR ONE COLUMN IN SIDE WALL OF THE ATLANTIC MUTUAL BUILDING.

ditions the battery should be installed and connected in such a manner as to consume all the excess current, taking the peaks of the elevator load requirements.

17. A double pressure tank system may be installed. The usual practice is to carry low pressure in the tanks running from one hundred and fifty to one hundred and seventy-five pounds for low pressure systems when the cars are few in number and centralized, and from seven hundred and fifty to eight hundred pounds when the plant is very large and scattered. A double pressure system would be very much more economical to operate, the low pressure tank operating under light service conditions at a pressure of from one hundred to one hundred and twenty-five pounds, and the high pressure tank under high service conditions at a pressure of from one hundred and seventy-five to two hundred pounds, an elastic system adapting itself to the work to be done.

18. There is a great deal of controversy as to the number of elevators to be installed. It depends absolutely on the class of building, and the nature of the business transacted therein and its location. Some engineers have compiled tables laying down specific floor areas to be served per elevator, but these rules do not always apply. The writer knows of one building having one elevator per eight thousand square feet of floor area which is under-elevatored, and of another close by, with an elevator to each sixteen thousand square feet of floor area which would have efficient service with fewer elevators.

19. Then, again, here is a list of buildings in most cases satisfactorily served which differ entirely from the above.

20. These buildings are all in the downtown business district of the city of New York, and are used as offices.

Building.	Stories.	Style.	Number of Cars.	Gross Sq. Ft. of Floor Area.
Broad Exchange	20	Hydraulic	18	25,860
Park Row	25	Electric	10	31,500
American Exchange Bank	16	Hydraulic	3	24,000
Bank of Commerce	19	Hydraulic	7	24,600
Wall Street Exchange	25	Hydraulic	10	22,500
Bishop	12	Electric	4	24,000
Beaver	15	Electric	4	21,000
Sixty Wall Street	26	Hydraulic	8	21,500

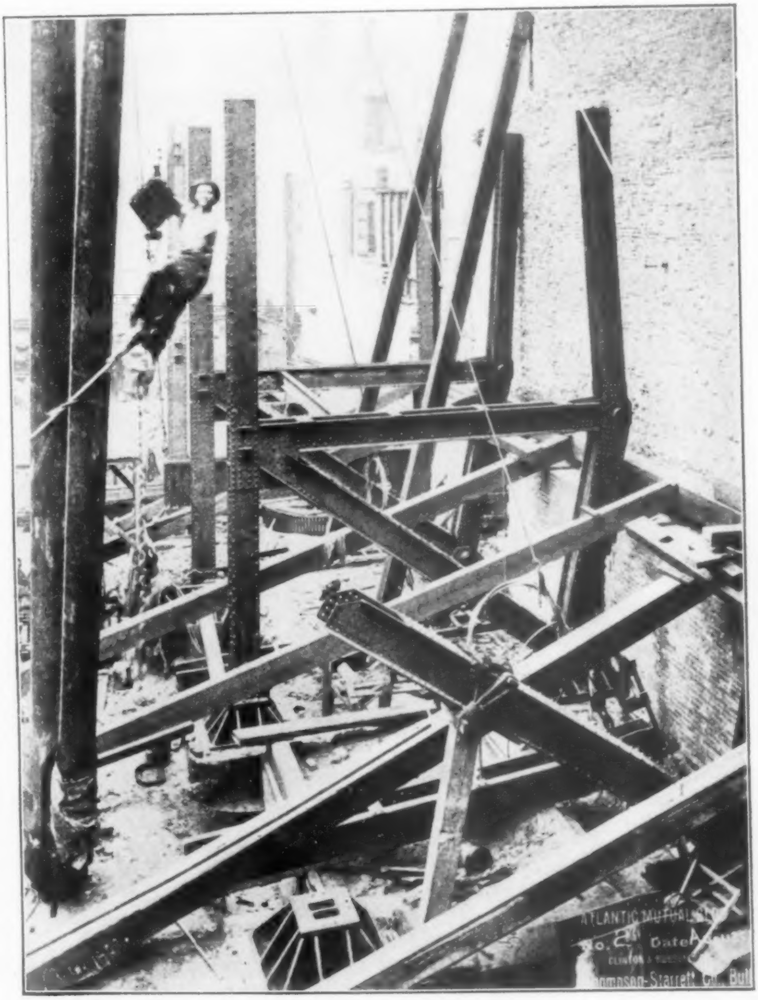
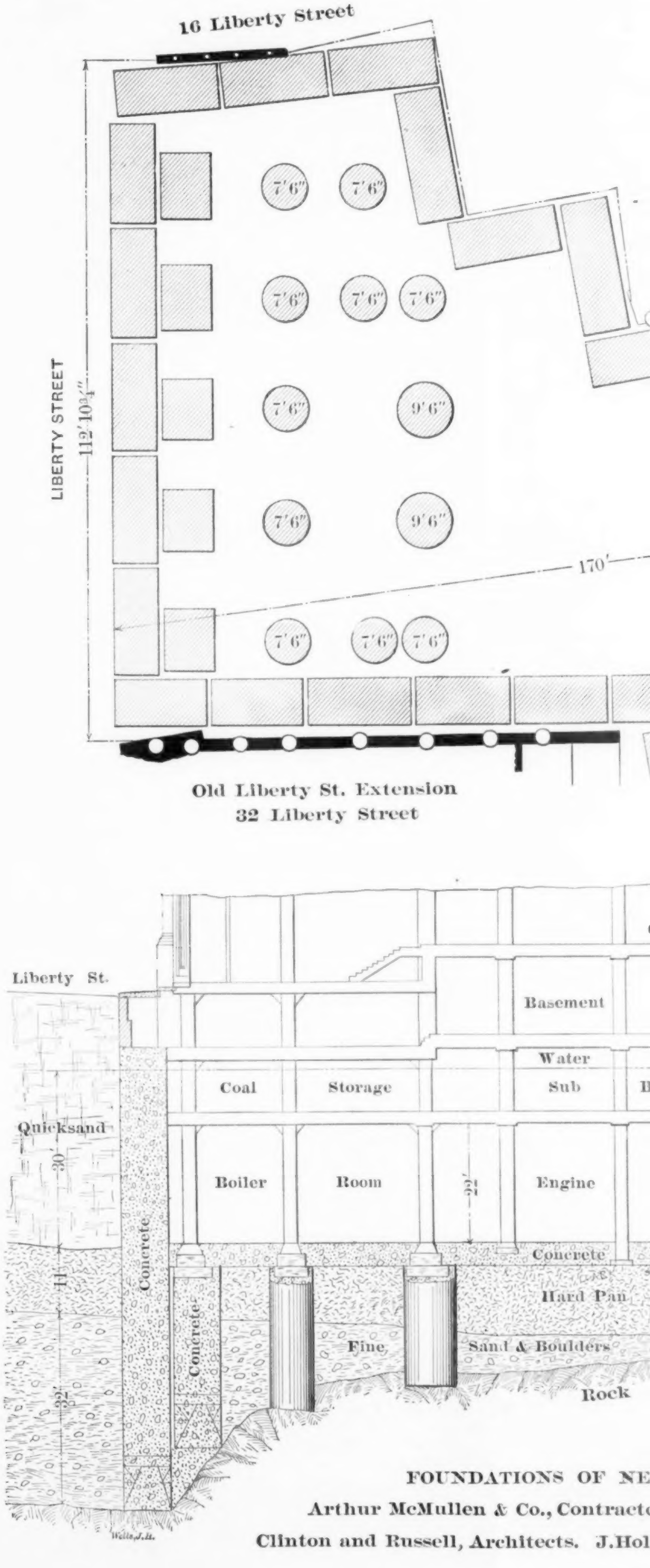


FIG. 347.—SAME AS FIG. 4, SHOWING COLUMNS ABOVE.

The Steam Plant.

21. This is a very large topic that cannot possibly be covered within the scope of this paper. The proper design of the steam plant is a matter of the utmost importance as on it depends most of the economies of the tall buildings. It may be divided into two parts, the boiler plant with its accessories, including the power piping, and the heating and ventilating systems. These have all been fully described in books and papers written by most competent, experienced engineers, and in practice they agree generally. In designing the boiler plant it is customary to allow about 50 per cent. reserve over the estimated peak loads. A modern boiler plant is usually equipped with such requirements as will produce economy. Superheaters of sufficient capacity to allow the steam produced by the boilers to receive a superheat of from 75 degrees to 100 degrees Fahr. have been used with excellent results, and, as a matter of course, all pressure piping and valving should be designed accordingly. Much difficulty has been experienced in pressure piping by the use of the ordinary flanged joint with copper gaskets, but the writer has had very satisfactory results with Van Stone joints, or joints somewhat similar in design, for piping exceeding 5 inches in diameter.

22. To avoid vibration in the building the steam and exhaust piping has to be very carefully installed, and is usually supported from the floor of the engine-room. The exhaust is carried above the roof and capped with a condenser head. This exhaust is connected into the heating system and also through the feed water heater. All drips are carried back to a drip tank and pumped back to the boiler with all returns through feed water filters. The equipment consists of the boilers, feed water heaters and pumps, filters, blow-off and drip tanks, house and fire pumps, back pressure and reducing pressure valves, hot water meters, recording pressure gauges, condensing and cooling coils, expansion and muffler tank, separators, pump governors, power, exhaust, heating and drip piping, engines and oiling systems, dynamos, ventilating, refrigerating, vacuum cleaning, electric wiring, switch board, motors, plumbing and water supply, and a multitude of appurtenances, all of which have to be carefully designed and placed according to the requirements of the building and space to be occupied.

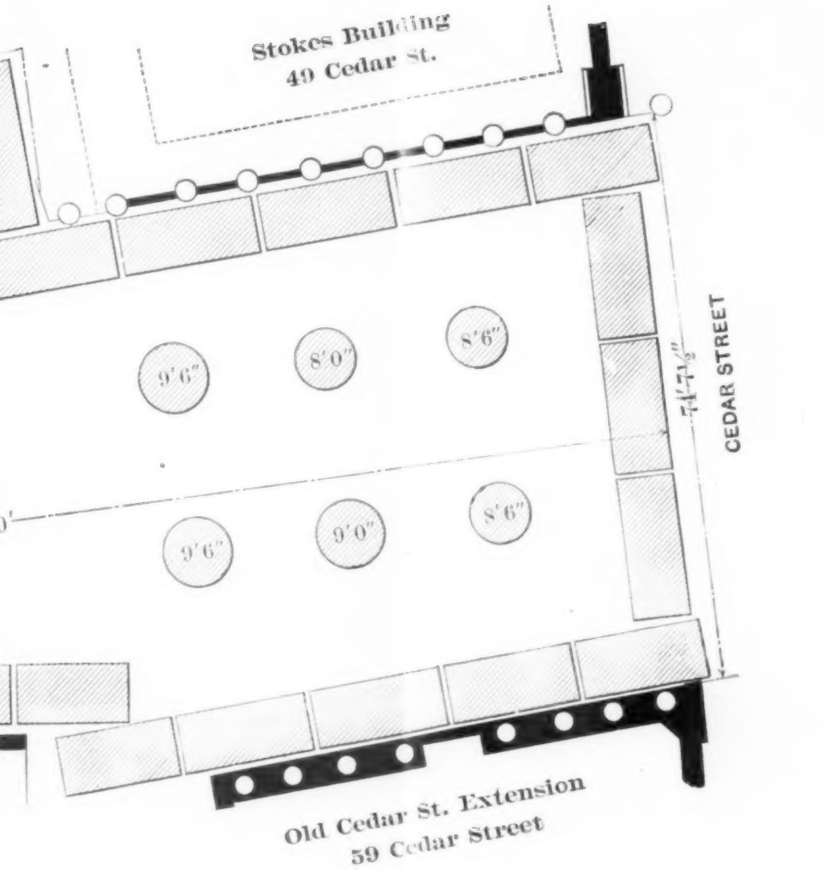


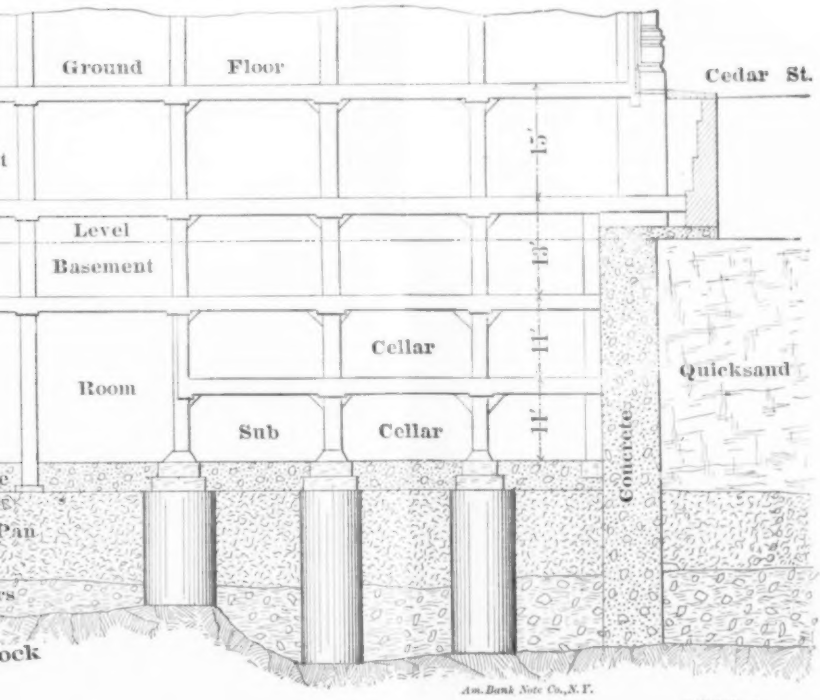
FOUNDATIONS OF NE

Arthur McMullen & Co., Contracto

Clinton and Russell, Architects. J.Hol

FIG. 34





1900-1.

F NEW MUTUAL LIFE BUILDING, NEW YORK.

tractors. T.Kennard Thomson, Engineer. Park Row Building

J.Hollis Wells, Engineer. Alfred Noble, Consulting Engineer.

FIG. 348.

PLAN

MUTUAL

Clinton & Russell
Architects.

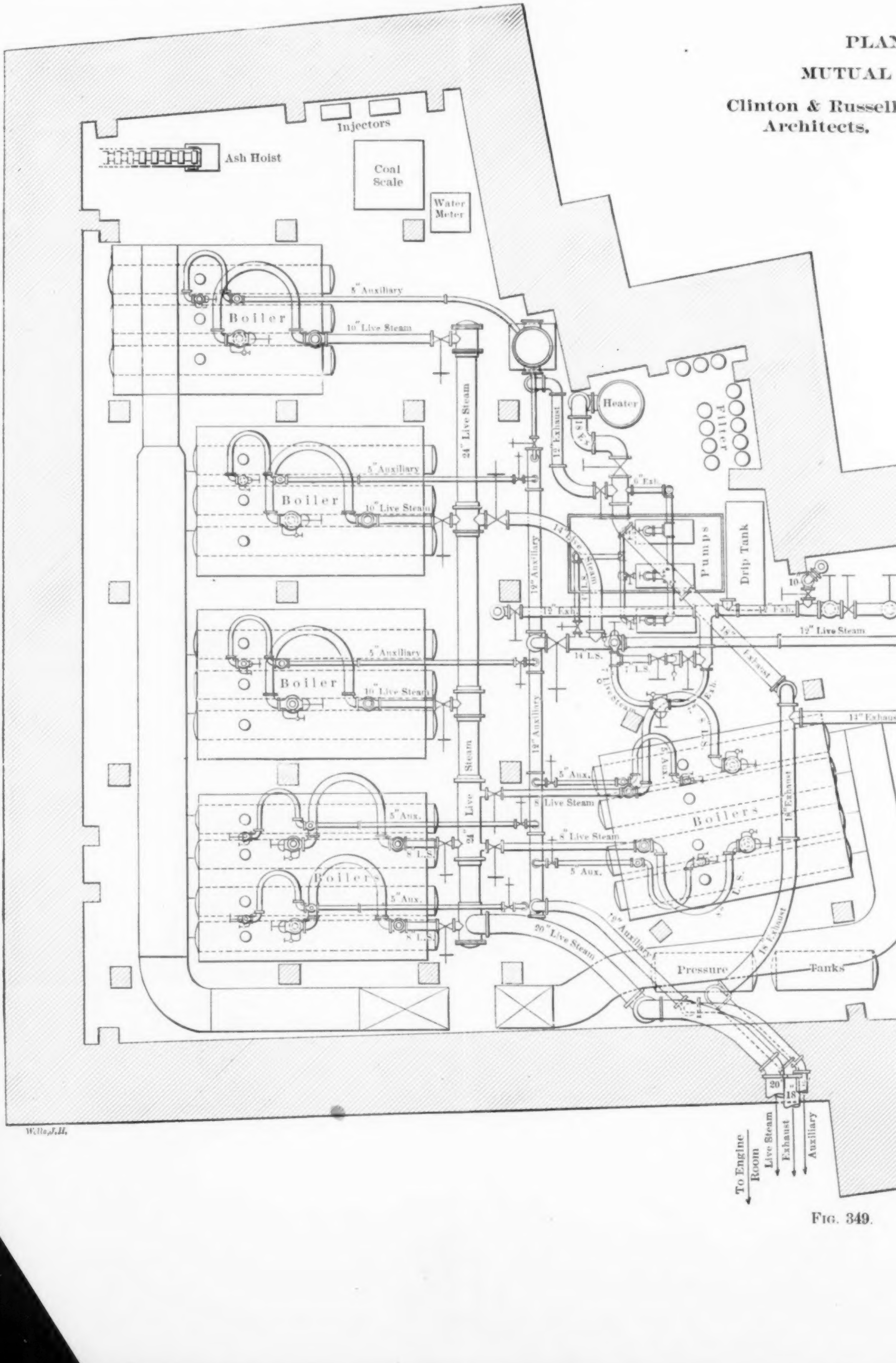


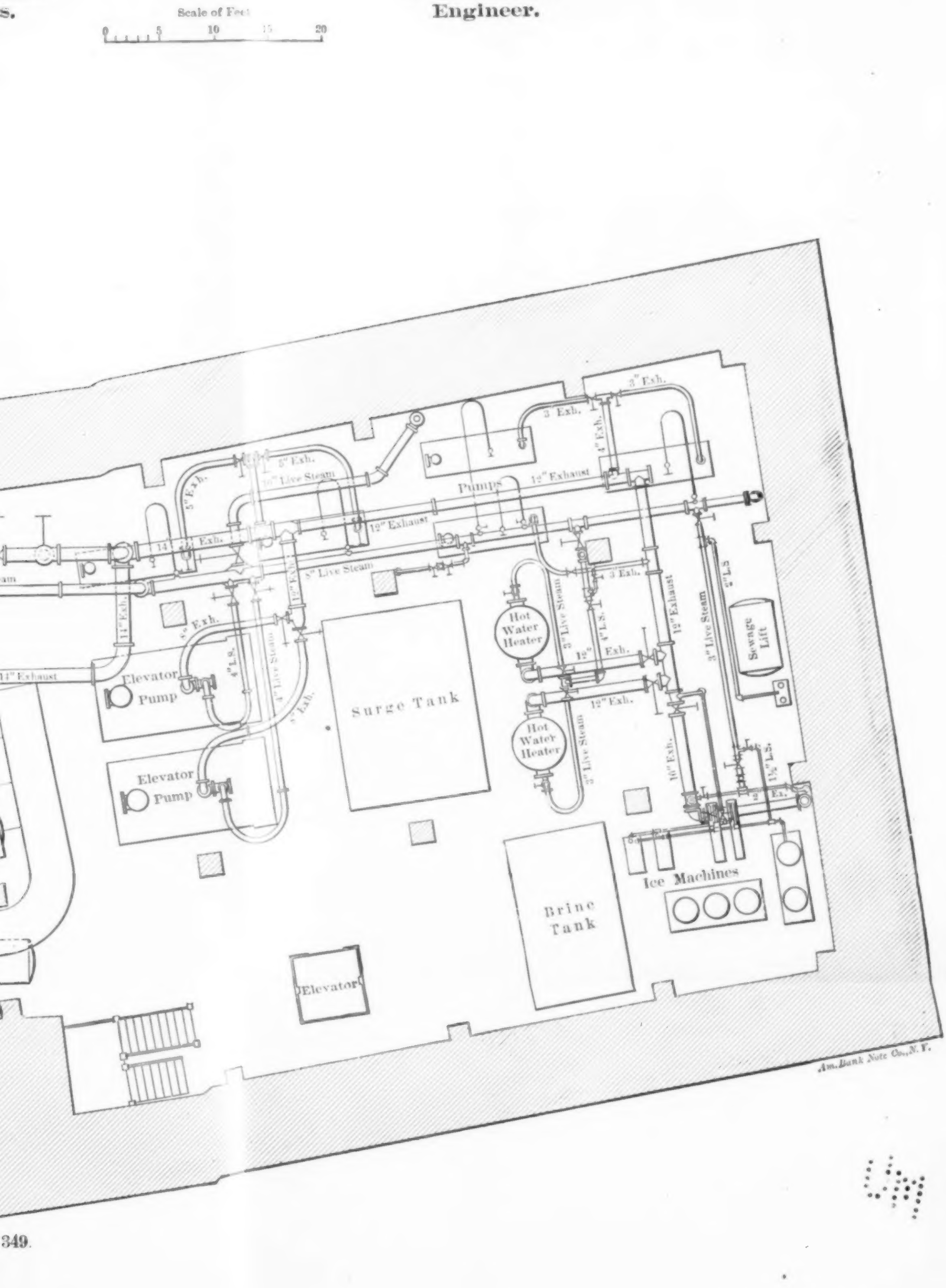
FIG. 349.

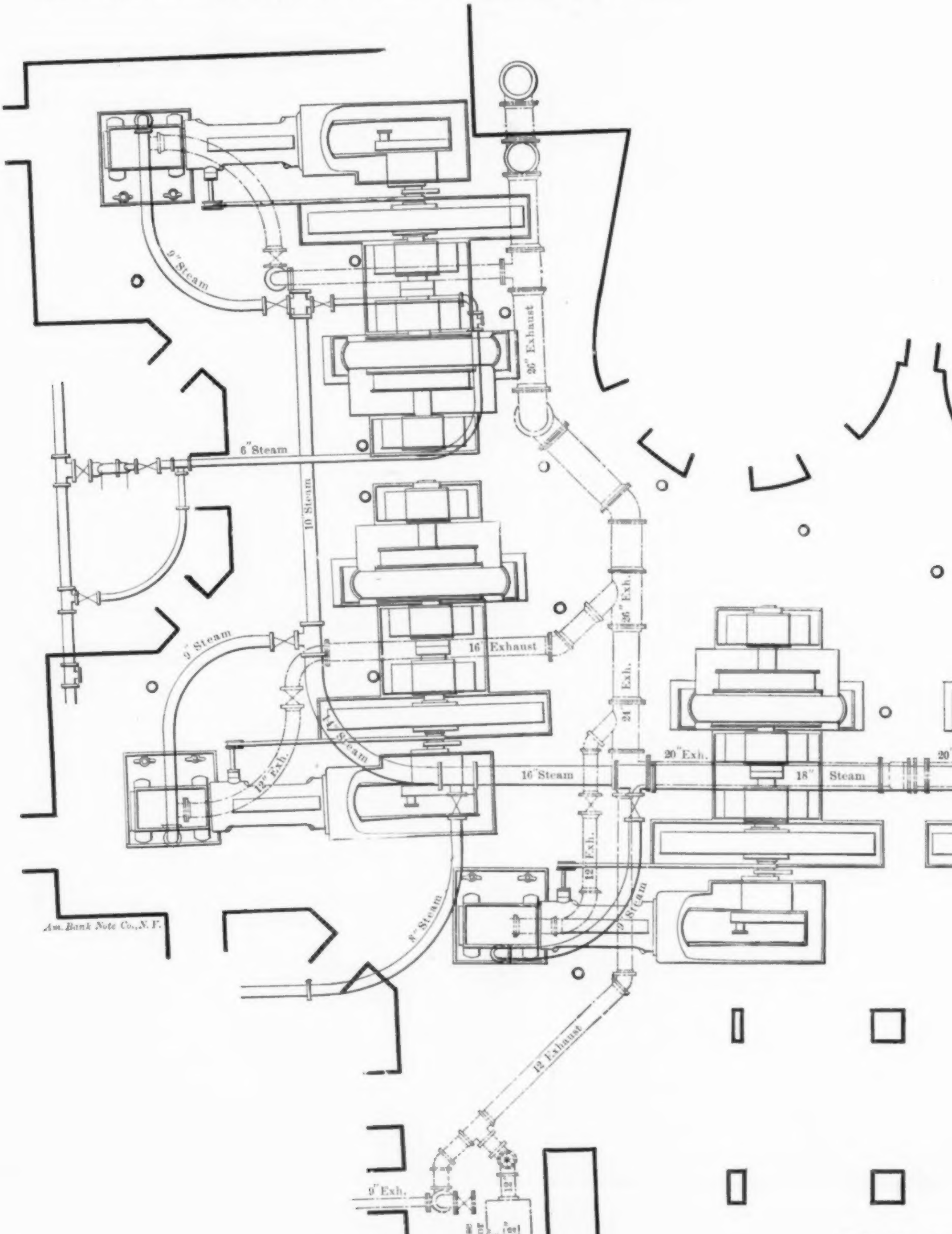
PLAN OF BOILER AND PUMP ROOM

UAL LIFE BUILDING, NEW YORK CITY.

ssell
s.

J. Hollis Wells
Engineer.





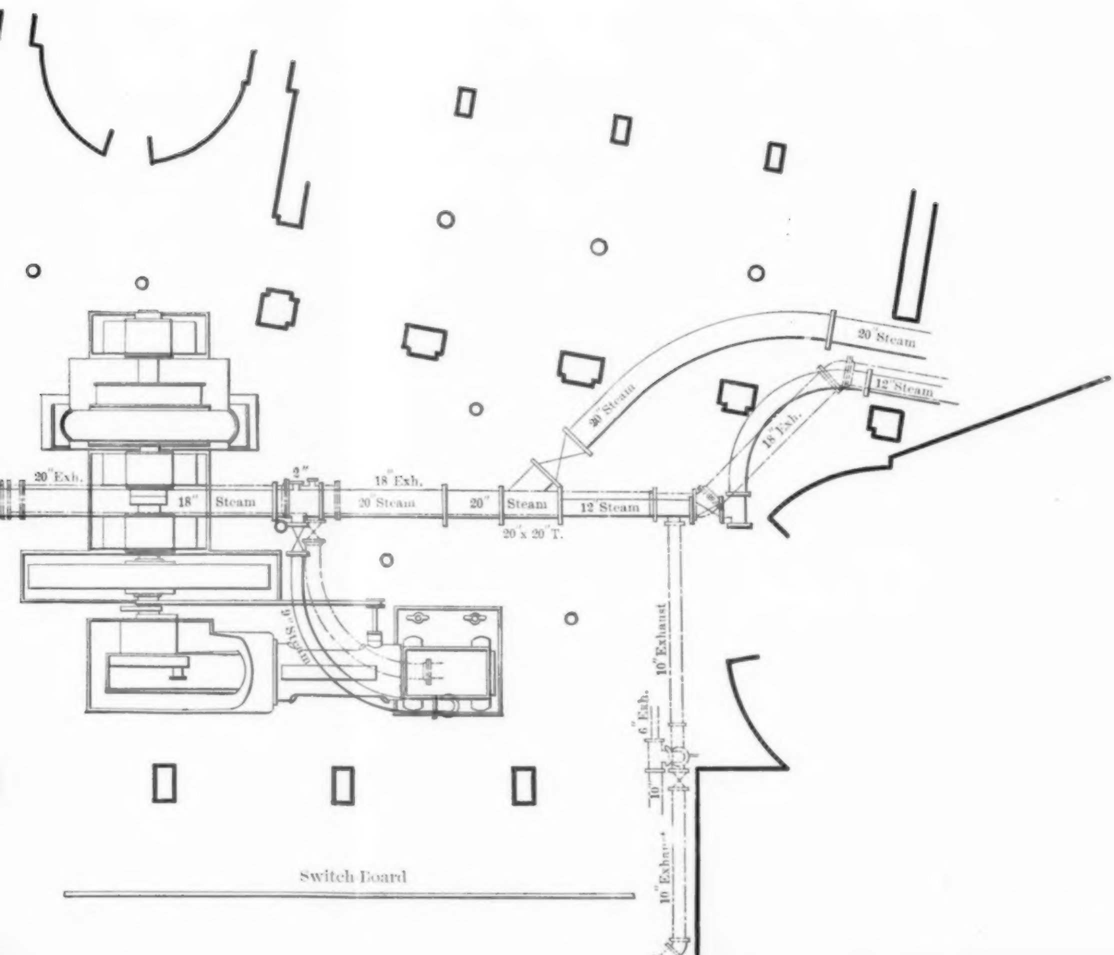
PLAN OF

MUTUAL I

Four 350 K-W.

Four 650 H.P. W

FIG.



S OF ENGINE ROOM

AL LIFE BUILDINGS

K-W., Bullock Generators

D.C. to

P. Watts-Campbell Engines.

FIG. 350.

23. In estimating and proportioning heating surface the method commonly used is that prescribed by law by the German Government in the design of heating plants in its public buildings, and is based on the loss in heat units per square foot of exposed surface transmitting heat to the outside air or buildings, as the case may be. On account of leakage it is also assumed that the air in the various rooms that are heated is changed once an hour, and that, therefore, a certain amount of heating surface is required to warm the air. Under certain conditions of location and exposure, and whether the building is to be heated continuously or intermittently, it is necessary to increase these estimates. In making the various increases the judgment and experience of the designing engineer must be used. In designing the mains and risers supplying steam to the radiators the sizes of pipes are determined by the amount of condensation of the radiating surfaces and the velocity of the steam flowing through the pipes. In mains this velocity is assumed to be from 60 to 80 feet per second, and in risers from 25 to 50 feet per second.

24. The mains and risers are either a two pipe system or a one pipe overhead system, depending somewhat upon circumstances. Both systems have their advocates, but either, if properly designed, gives satisfactory results. An exhausting apparatus is often used in connection with the air lines, and the system is then run nearly at atmospheric pressure, thereby relieving the engines from back-pressure, a source of economy. It is a mistake, however, to reduce the size of the risers where it is intended to install this apparatus, for the heating system should be designed to work under gravity in case of the breakdown of the exhauster. There are several other devices of a similar sort often used, but the writer's experience with the straight-out gravity system has been most satisfactory. The other devices can be added, but the system will always work.

25. The artificial heating of a building is a problem which usually causes much thought and perplexity, and the solution is often very far from giving satisfaction. It seems to be simple enough in theory, but there are so many varying influences in practice to be guarded against that, however well arranged, no plans seem exactly to meet the conditions at all times. It is usually agreed in contracts for heating that the heating plant shall be capable of keeping the several apartments at a tempera-

ture of 70 degrees when the thermometer in the open air stands from zero to 10 degrees below. The apparatus, therefore, that fulfills this condition will be apt to overheat the rooms when the outside temperature is above the maximum, compelling the occupants to give strict and almost constant attention to the shutting off of the heat sources. Man is fallible, and often negligent; nerves and senses are slow to feel gradual changes in temperature until a degree is reached that is far from normal; consequently, rooms regulated by human hand are usually too hot or too cold, and often have sudden variations of from 5 to 10 degrees.

26. A cold wind will often make it extremely difficult to get one side of the building warm, while the others suffer from an excessive heat. Frequent and sudden variations in temperature, excessive heat alternating with chilling cold, are unhealthful and debilitating, and it is a well-known fact that a great deal of sickness is directly due to inefficient or absolute absence of control of the supply of artificial heat. In order to prevent such excessive variations, preserve a uniform temperature, to distribute the heat where and as it is wanted, and economize fuel by preventing waste of heat, heat regulation is often employed, and to get proper results, the temperature of each room in the building is controlled independently of the others.

27. Such a system will insure a uniform and reasonable temperature at all times, never varying over one or two degrees, regardless of the variations of outside temperature and without any thought or attention whatever from the occupants, thereby decreasing the care and trouble incident to heating plants, entirely removing the necessity of opening or closing steam valves or registers, and promoting health and comfort. By a simple device the degree at which regulation occurs can be varied, so that different rooms can be kept at different temperatures or changed as frequently as desired. By the same system, where air for ventilating purposes is supplied by a fan system, this air can be kept uniform and at any temperature desired.

28. There seems to be a general complaint that office buildings are, as a rule, too hot for the personal comfort of the occupants, and the reason is simply that on account of the necessity of having to supply radiators large enough to meet the maximum requirements, and the inattention that is usually given to such radiators, it is only natural that such results should follow, because the occupants are usually busy with their regular duties, and the at-

tention is not called to the overheating of the rooms until the temperature has reached a most uncomfortable point. Then the heat is shut off and the reverse takes place for the same reasons. It is usually customary to open the windows in order to reduce the heat, and perhaps to a certain extent to regulate it by admitting cooler air from the outside. This, of course, results in draughts, which are not only disagreeable but dangerous to health. It is no uncommon sight in moderate weather to observe that a very large per cent. of the windows in office buildings are opened in the attempt to overcome the disagreeable condition incident to the heating plant, thus demonstrating the necessity of some method of regulating the temperature.

29. It is an almost self-evident proposition in the case of tall buildings that, where the heat sources are controlled by a thermostatic system, at least 50 per cent. of the offices will have their heat shut off most of the time, which is proof that a reduction or saving of that amount of heat will be effected. As it is usually customary to heat such buildings with the exhaust steam from the engines used for the electric or elevator plants, little actual saving in the cost of heat can be effected. This, however, would be quite an item if the steam was generated directly for heating purposes. The amount of saving in the cost of heating perhaps should not be considered as important as the health and comfort of persons renting the various offices, and for whose comfort the owners of buildings usually furnish such modern improvements as will render them desirable for occupancy and keep them fully occupied.

30. Artificial ventilation by means of fans electrically driven is often used on the lower stories. One tall building recently erected in New York has every room heated and ventilated by means of an indirect system operated by fans. This is an extraordinary case, however; it is usually impracticable on account of the cost of installation and the amount of room taken up by the fans, heaters and ducts. In the New York Stock Exchange the air is warmed in winter, and by means of an immense refrigerating plant cooled in summer. These are all special methods, however, and the great bulk of the heating is done by the direct method; that is, radiators under windows in the rooms.

31. In many buildings there are refrigerating plants installed both for cooling the drinking water and for making ice which is sold to the tenants.

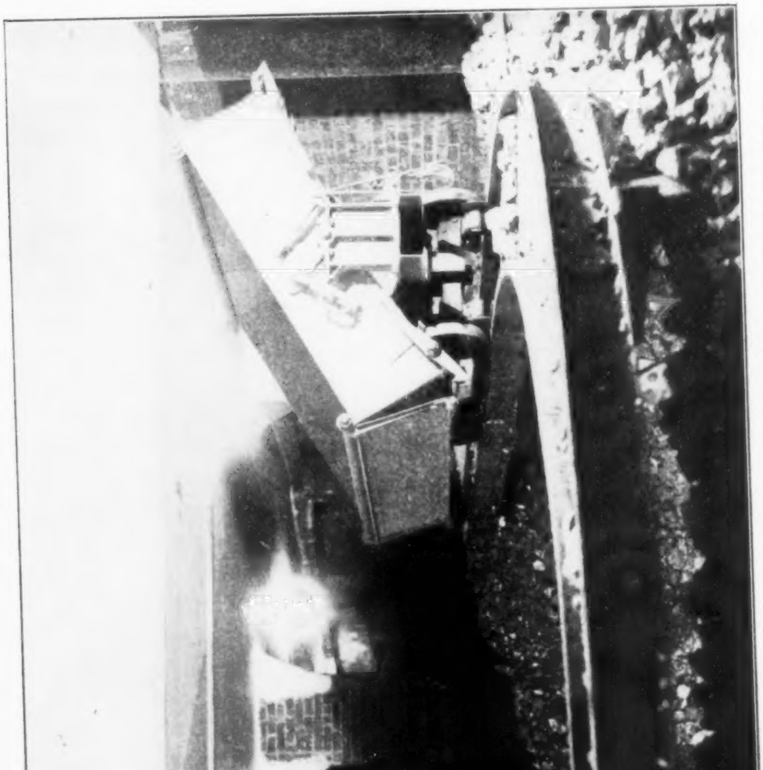


FIG. 351.—COAL BUNKER IN THE BASEMENT OF THE MUTUAL LIFE INSURANCE CO., 30 FEET BELOW THE CURB.

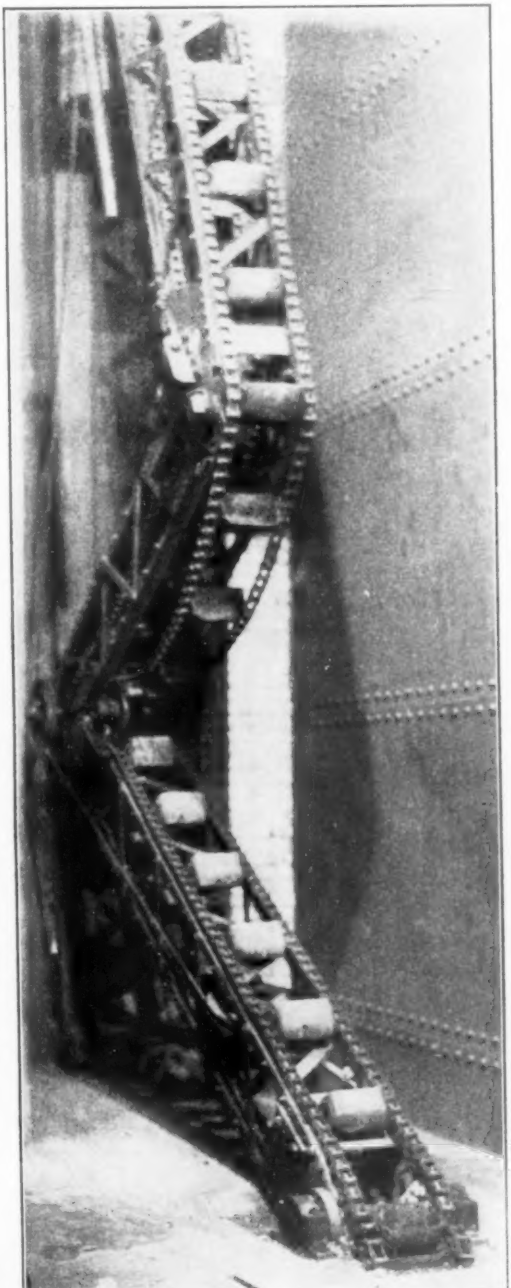


FIG. 352.—ASH HOIST FROM CELLAR TO SIDEWALK, MUTUAL LIFE INSURANCE CO. OPERATED BY 10-H.P. ELECTRIC MOTOR.

32. Cleaning systems have recently come into use, and have given much satisfaction. These are air systems operating either under vacuum or compression.

The Electric Plant and Wiring.

33. The engineer who is at the head of this department has his work cut out for him. It is the most complex and difficult portion of the design and requires a large amount of technical education and experience. Practice changes continually, and the science has advanced so rapidly that an up-to-date electrical engineer has to give more time to his profession than does the engineer in any other department. In laying this portion of the work before you the writer will take a specific case which will illustrate the method of design. Let it be understood, however, that the figures used are an average, and that they are the results of tests and records made in perhaps fifty of the tall buildings in New York City. The building in question is designed for banking rooms on the first and second floors and above offices. Three boilers of 350 horse power each are located in the cellar and the plant in the basement. The electric portion of this plant consists of two 125 kilowatts, one 100 kilowatts and one 50 kilowatts generators, two fan motors of 15 horse power each, and engine plant of sufficient horse power to operate the entire electrical installation. These engines will be of the four-valve tandem compound type and the dynamos direct current, 120-volt machines. The total floor area to be lighted is 196,700 square feet, and taking the average rate at one light for each 37 square feet of floor area, the number of lights as laid out on the plans is 5,316 plus 250 lights allowed for decorative effect in banking rooms, the total number of lights to be wired for is 5,566. Lamps are guaranteed at 50 Watts per 16 candle power, but this is when they are new; we, therefore, assume an average of 55 Watts; this, therefore, means that with all the lights burning the total output would be 306 kilowatts. From experience we find the average working loads to be as follows:

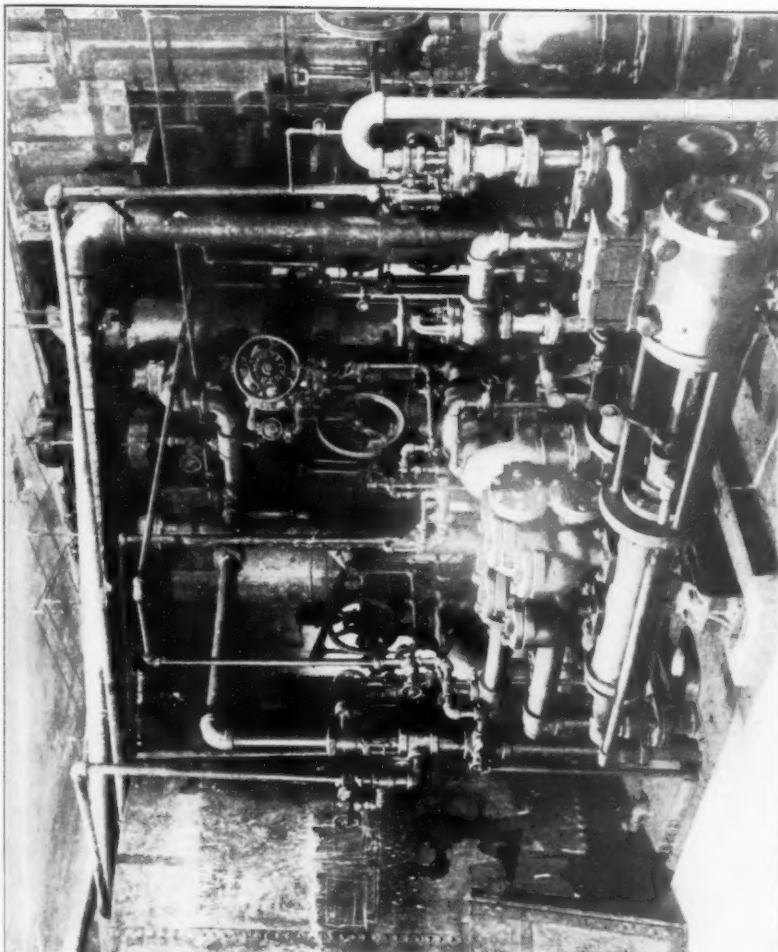
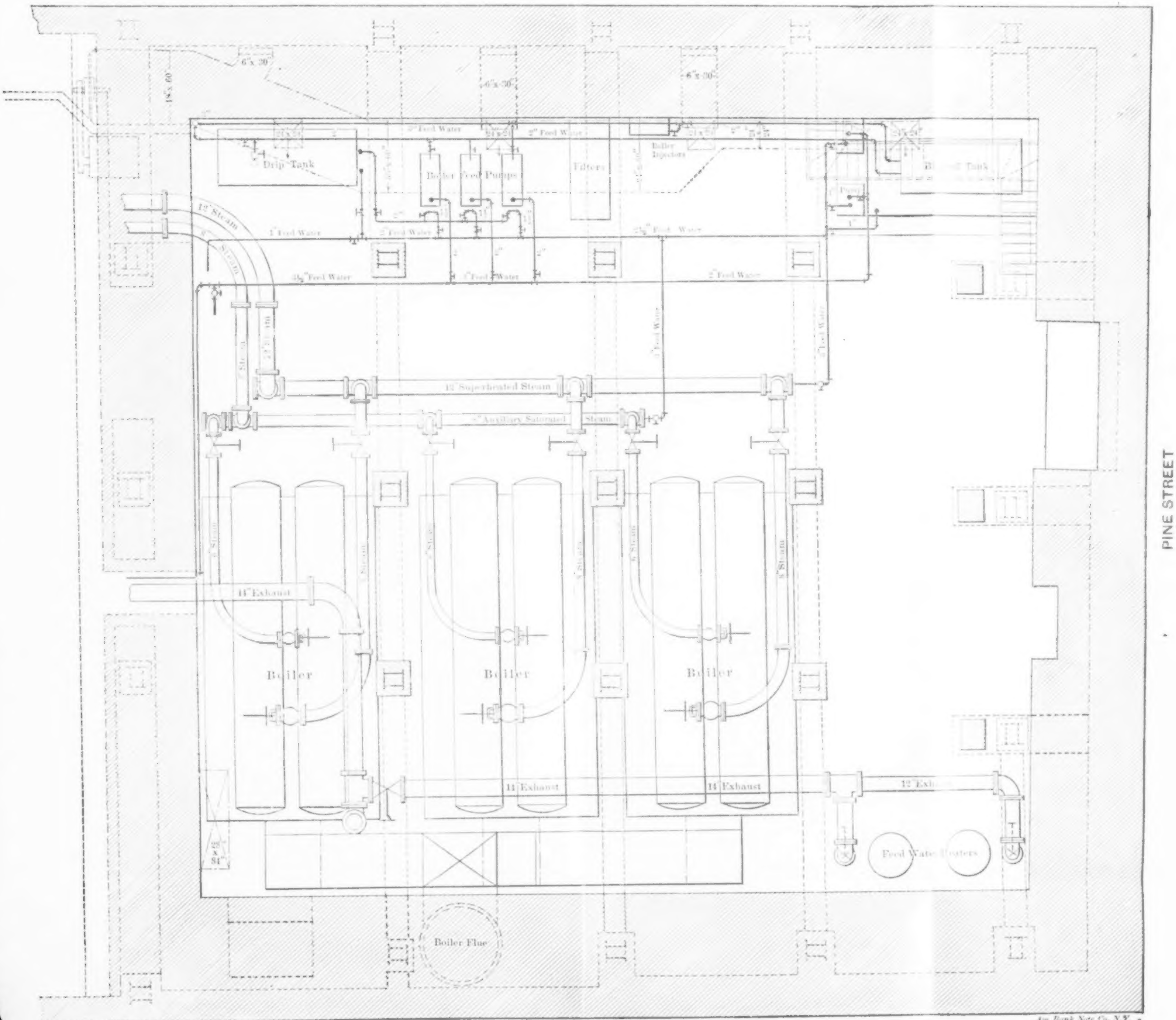


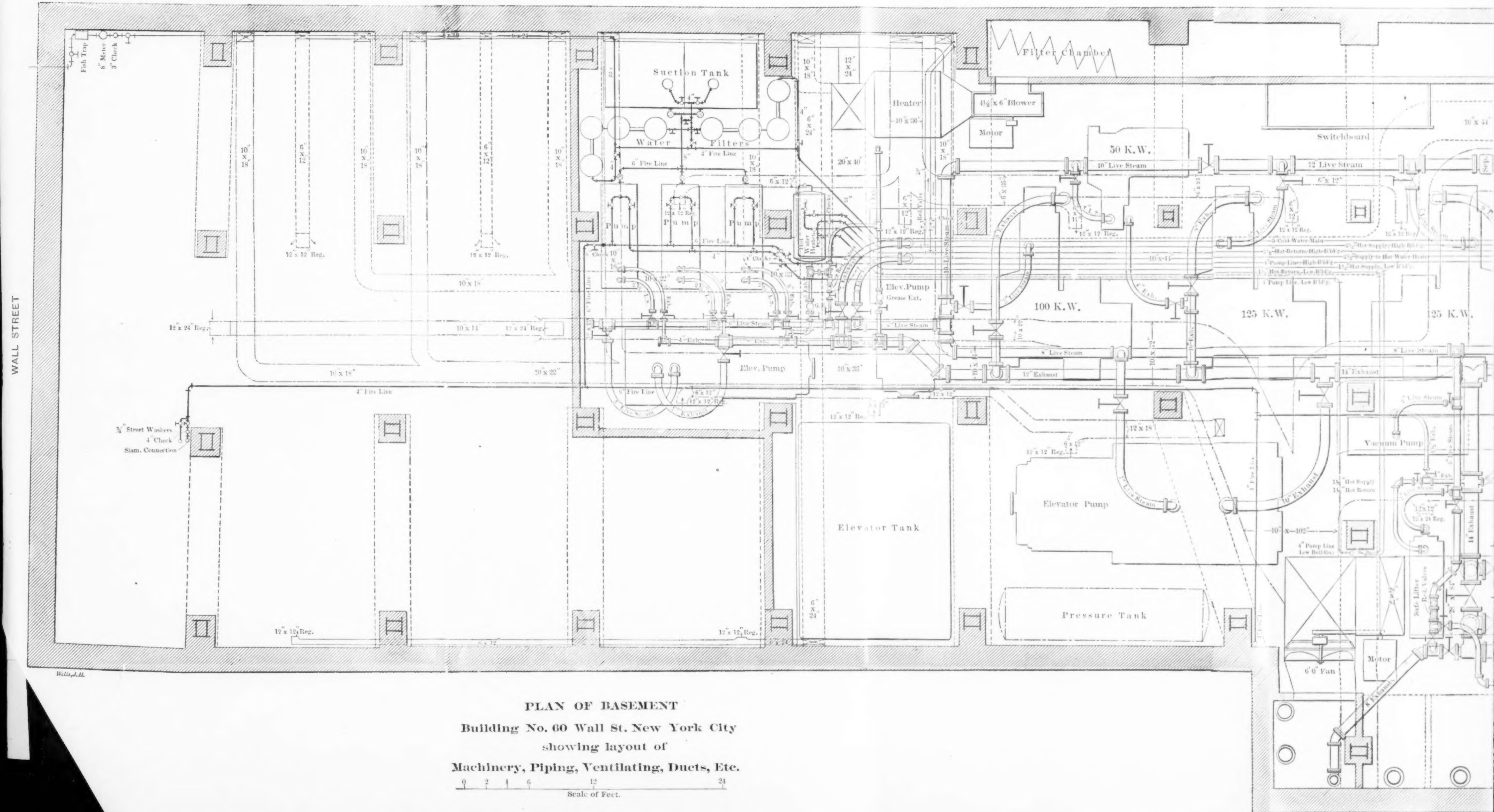
FIG. 353.—SAFE LIFTING PUMPS AND BOTTOM OF ELEVATOR CYLINDERS IN CELLAR OF THE MUTUAL LIFE INSURANCE COMPANY BUILDING.



PLAN OF BOILER ROOM
in Sub-Basement of
Building at No. 60 Wall St. New York City.

SCALE OF FEET.

FIG. 354.



PLAN OF BASEMENT

Building No. 60 Wall St. New York City

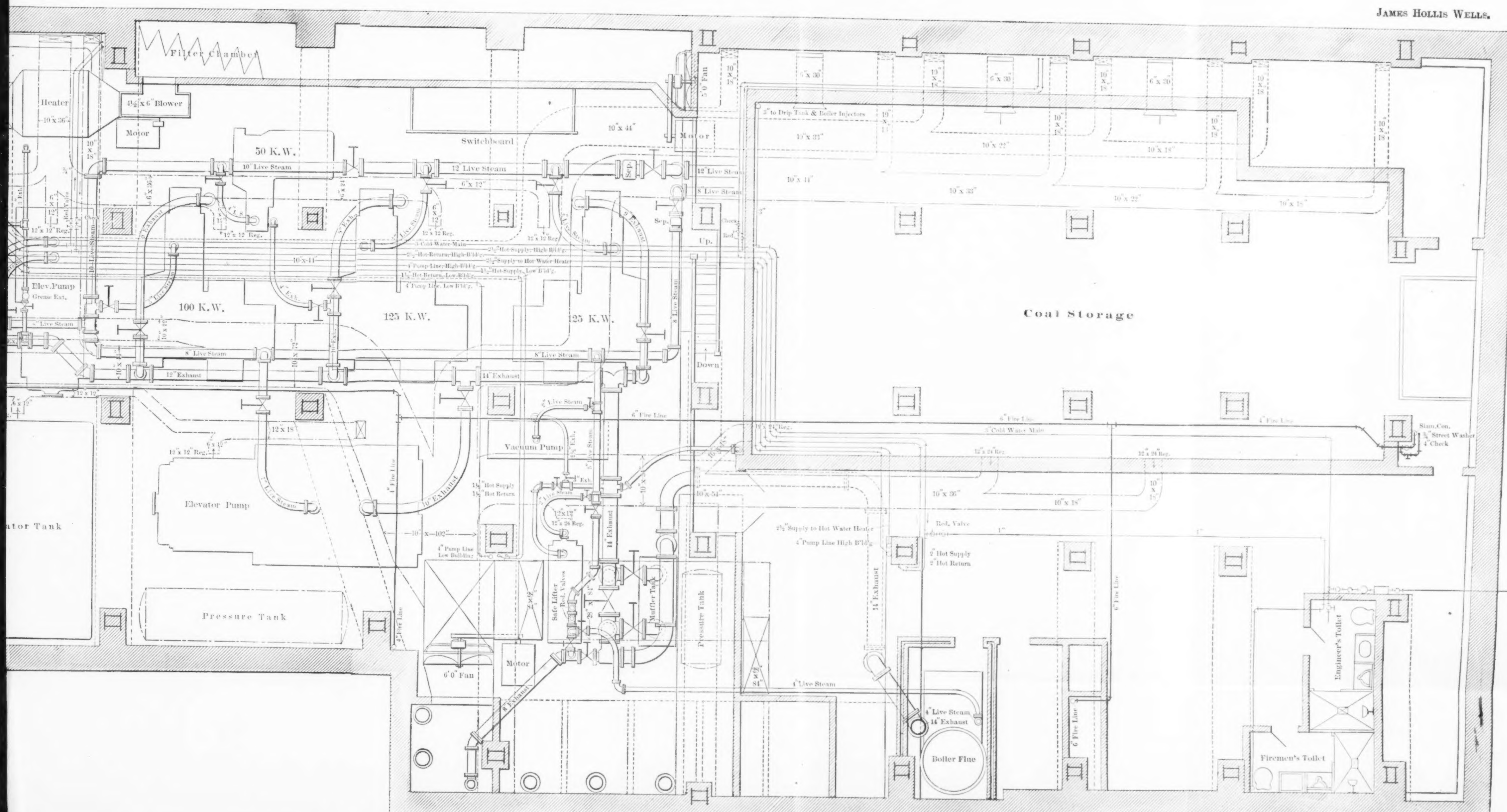
showing layout of

Machinery, Piping, Ventilating, Ducts, Etc.

0 2 4 6 12 24

Scale of Feet.

FIG. 355.



Filter Chamber
Heater
4½ x 6" Blower
Motor
Switchboard
50 K.W.
100 K.W.
125 K.W.
125 K.W.
10" Live Steam
12" Live Steam
12" Live Steam
8" Live Steam
8" Live Steam
12" Exhaust
14" Exhaust
Motor
Coal Storage
Elev. Pump
Grease Ext.
Elevator Pump
Vacuum Pump
Pressure Tank
Pressure Tank
Muffler Tank
Safe Lifter
Red. Valves
6'0" Fan
Motor
Boiler Flue
Engineer's Toilet
Firemen's Toilet
Siam. Con.
Street Washer
4" Check
6" Fire Line
4" Fire Line
Cold Water Main
2½" Supply to Hot Water Heater
4" Pump Line High B'ld'g
2" Hot Supply
2" Hot Return
Red. Valve
4" Live Steam
14" Exhaust
Up.
Down
PINE STREET

FIG. 355.

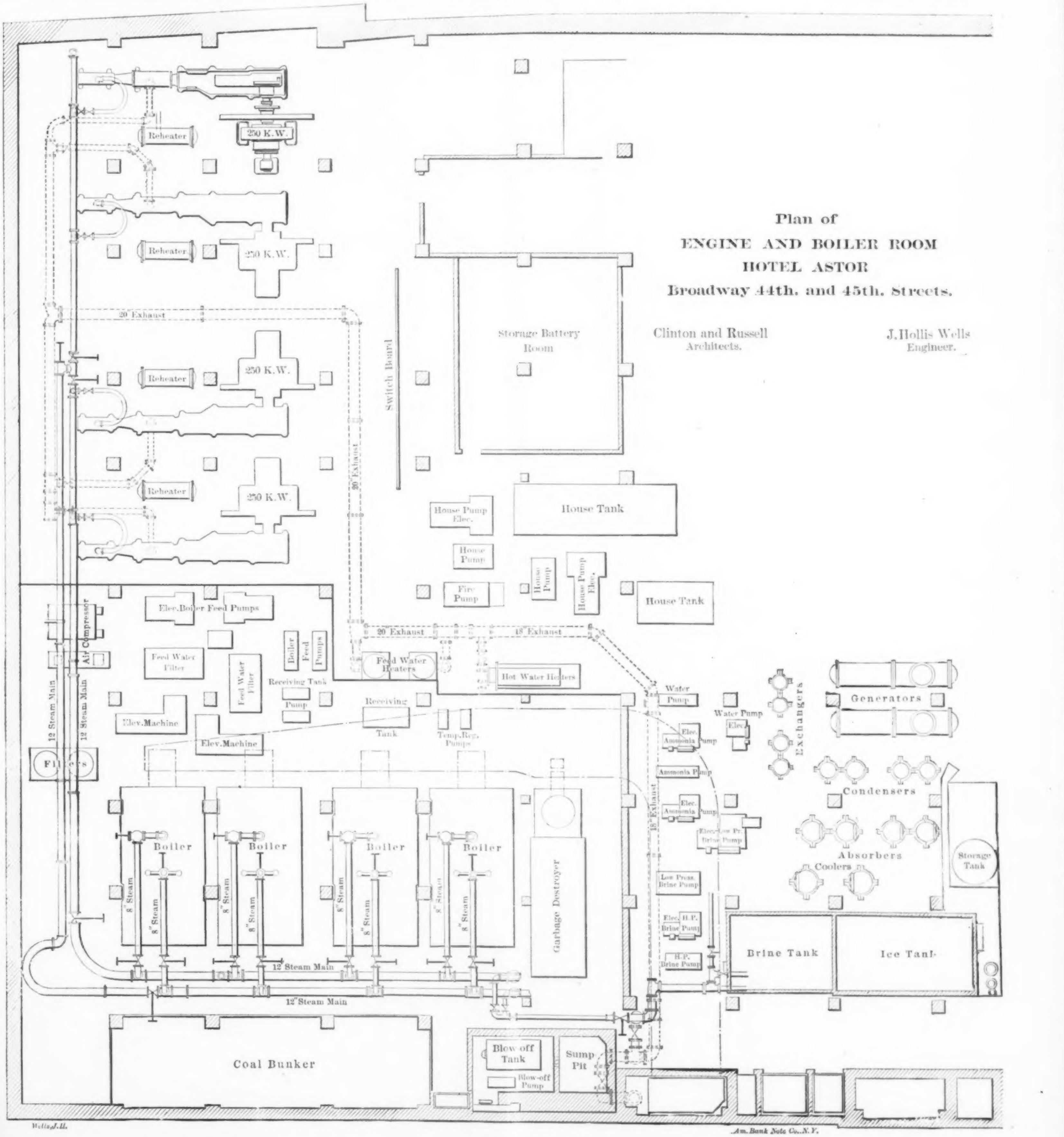

Plan of
ENGINE AND BOILER ROOM
HOTEL ASTOR
Broadway 44th. and 45th. Streets.
Clinton and Russell
Architects.
J. Hollis Wells
Engineer.
250 K.W.
Reheater
20 Exhaust
18 Exhaust
Switch Board
Storage Battery Room
House Tank
House Pump Elec.
House Pump
Fire Pump
Elec. Boiler Feed Pumps
Air Compressor
Feed Water Filter
Boiler Feed Pumps
Receiving Tank
Pump
Feed Water Heaters
Hot Water Heaters
Elev. Machine
Temp. Reg. Pumps
Water Pump
Elec.
Exchangers
Generators
Ammonia Pump
Elec. Ammonia Pump
Condensers
Absorbers
Coolers
Storage Tank
Low Pres. Brine Pump
H.P. Brine Pump
Brine Tank
Ice Tank
12 Steam Main
Filters
Boiler
8" Steam
Garbage Destroyer
12" Steam Main
Coal Bunker
Blow off Tank
Blow-off Pump
Sump Pit
Wells, J.H.
Am. Bank Note Co., N.Y.

FIG. 356.

TABLE SHOWING KILO-WATT HOURS OF LIGHTING LOAD.

WINTER SCHEDULE:

Months.	Days.	Sundays.	Holidays.	Working days.	7 A. M. to 8 A. M.				8 A. M. to 3 P. M.				3 P. M. to 6 P. M.				6 P. M. to 7 P. M.				Total K.-W. hours per month.
					Per cent. load.	Equiv. K.-W.	No. hours.	Total K.-W. hours.	Per cent. load.	Equiv. K.-W.	No. hours.	Total K.-W. hours.	Per cent. load.	Equiv. K.-W.	No. hours.	Total K.-W. hours.	Per cent. load.	Equiv. K.-W.	No. hours.	Total K.-W. hours.	
January	31	4	1	26	20	61	1	1,586	25	76	7	13,832	60	183	3	14,274	50	153	1	3,978	33,670
February	28	4	2	22	18	55	1	1,210	24	74	7	11,396	50	153	3	10,098	40	122	1	2,684	25,388
March	31	4	0	27	17	52	1	1,404	23	70	7	13,230	33	101	3	8,181	20	61	1	1,647	24,462
October...... ..	31	4	0	27	17	52	1	1,404	24	74	7	13,986	34	104	3	8,424	21	64	1	1,728	25,542
November	30	4	2	24	19	58	1	1,392	30	92	7	15,456	60	183	3	13,176	50	153	1	3,672	33,696
December.......	31	5	1	25	21	64	1	1,600	34	104	7	18,200	70	214	3	16,050	60	183	1	4,575	40,425
SUMMER SCHEDULE:					7 A. M. to 8 A. M.				8 A. M. to 4 P. M.				4 P. M. to 6 P. M.				6 P. M. to 7 P. M.				
April...........	30	5	0	25	14	43	1	1,075	21	64	8	12,800	30	91	2	4,550	20	61	1	1,525	19,950
May............	31	4	1	26	14	43	1	1,101	19	58	8	12,064	29	89	2	4,628	19	58	1	1,508	19,301
June	30	4	0	26	11	33	1	858	18	55	8	11,440	28	86	2	4,472	18	55	1	1,430	18,200
July............	31	5	1	25	10	30	1	750	17	52	8	10,400	27	83	2	4,150	17	52	1	1,300	16,600
August	31	4	0	27	10	30	1	810	17	52	8	11,232	27	83	2	4,482	17	52	1	1,404	17,928
September	30	5	1	24	12	37	1	888	19	58	8	11,136	29	89	2	4,272	19	58	1	1,392	17,688
													Total day load per year..................								292,850

Total day load per year	292,850
Average night loads at 5 p. c., or 16 K. W., 365 nights, 12 hours per night per year	70,080
Average Sunday and holiday loads at 10 K. W. 61 days, 12 hours per day per year	7,320
Motor K. W. hours per year	60,800
Grand total K. W. hours	431,050

38. Assuming, therefore, the total output of the machines through the husbars of the switchboard to be 431,050 klowatts hours per year, the engines would generate

$$1.6 \times 431{,}050 = 689{,}680 \text{ horse-power hours,}$$

and the boilers would generate

$$1.8 \times 431{,}050 = 775{,}890 \text{ horse-power hours,}$$

39. Assuming an engine guarantee of 24 pounds of steam per horse-power per hour and a direct evaporation of 8 pounds per pound of coal, then the total coal consumption will be 1,164 tons, and the water consumption 297,942 cubic feet. It is safe to assume that only 50 per cent. of this water is wasted and that the remainder is returned to the boilers.

40. Taking the cost of a good grade of coal at \$3.75 per ton, water at 10 cents per 100 cubic feet, oil and waste at \$400, interest and depreciation of \$3,000, ash removal at \$218.25, 5 per cent. of the cost of the coal, and charging one-half the cost of the force in the fire and engine rooms to this account or \$2,500 would make the total cost of operating the electric plant \$10,632, or approximately 2½ cents per kilowatt hour.

41. All of the data herein contained are from actual practice and do not wholly agree with theory, but as we have before stated, there are so many elements entering into the problem that sooner than base everything on pure theory we necessarily let the results of practice govern. In the design and construction of dynamos the trade seems to agree, but there is the great diversity of practice among high-speed engine builders. One will use a 1,200 pound fly-wheel on an 85 horse-power engine, and another calls for 2,000 pounds. Sizes of shafts, lengths of bearings, and in fact nearly all parts vary so materially that one wonders why there is not some absolute rule adopted by them. The designs are so similar that there is no reason for the great dissimilarity in sizes, unless it can be a commercial reason.

42. We have not attempted to more than touch upon the fact that there must necessarily be fire equipment, and well designed systems of water supply in tall buildings, and that large pumping

plants are also an essential part of the equipment. Reservoirs are established both in the cellar and on the roof, connections made on all floors, and in case of fire with Siamese or twin connections with the street. Water pressure must be reduced in sections by means of reducing pressure valves and great care exercised to avoid leaks, undue strains or water-hammer in the pipes. In addition, the building is equipped with interior telephone, ticker, messenger and bell wiring, mail chutes, ventilating screens, and in fact innumerable devices for comfort and convenience that aid to rent it. The very best of talent is studying every day to improve, and it is the aim of the designer to make each building superior to its predecessor, until it seems as though now we must have reached the very acme of success, and to-day the wonder of the world is the American Tall Building. It is safely constructed, comfortable and sanitary, a model in every respect and a monument to the American Architect and Engineer.

43. Great minds and many are studying the question under discussion. Much credit is due to the American Engineer and to the inventor for his work. The field is large; there is still much to be done, but we may safely leave this to those who are now making the subject a specialty. More and more engineers are entering this field, and the demand for their services becomes greater as time goes on. Contractors recognize the value of their services on the work, and it is needless to say that as in all other great works, the services of the engineer is a large factor in a successful undertaking. By no one is he more appreciated than by the architect whom he works in conjunction with; who creates the idea and leaves to him the large question of structural and mechanical design.

DISCUSSION.

Mr. William H. Bryan.—Mr. Wells presents much interesting and valuable data. Taking up his design of plant (paragraphs 33, 34 and 35), I have followed a different plan in my own practice. For a maximum load of 257 kilowatts, I should install three 100 kilowatt units instead of two 125, one 100, and one 50. Such peaks continue but a short period, and modern units are designed to carry overloads of $\frac{1}{3}$ to $\frac{1}{2}$. Two units would, therefore, take care of the peak, leaving the third in reserve. The resulting plant would be much simpler, cheaper to install and to

operate all units and their repair parts would be interchangeable, the floor plan would be better, the plant would occupy less space, while the reserve capacity would be ample to meet all requirements. If extreme economy in first cost is necessary only one engine need be compound. The others being reserve engines, used only occasionally, could be simple.

It is true that the four units proposed could, theoretically, be adapted to varying loads with greater fuel economy. In practice, however, operating engineers will not change their units over for small changes of load, and they prefer, furthermore, to always have a surplus of power in motion to take care of a possible sudden increase of load.

The table in paragraph 33 is not altogether clear, particularly the third and fourth lines. The attempt to predict the hours and amount of load is laudable; but, in my judgment, largely futile. No two buildings are alike. Everything depends upon the building, its construction, natural light, and the business carried on by the occupants. Mr. Wells' units suit his assumed loads, but would probably meet the actual requirements of any particular building no better than mine. Furthermore, some economy is always possible by varying the steam pressure to suit the actual loads.

Mr. Wells gives no rules for computing boiler power, but his ratio of 1.8 boiler horse-power per kilowatt of dynamo load seems higher than necessary. Accepting his engine water rate of 24 pounds per indicated horse-power hour (paragraph 39), the standard boiler unit of 30 pounds of water from feed of 100 degrees Fahr. into steam of 70 pounds gauge pressure, appears more than ample and would take care of all auxillaries and condensation losses. I should consider a ratio of 1.5 boiler horse-power per kilowatt sufficient. On this basis the peak load of 257 kilowatt would demand 385 rated boiler horse-power. The modern boiler is designed to be overworked from $\frac{1}{3}$ to $\frac{1}{2}$ continuously during peak loads, so that the above capacity could easily be generated with boilers of 289 horse-power rated capacity. I should design for such a plant three boilers of 150 rated horse-power each, two to do the maximum work and one for reserve. This computation does not, of course, take into account the steam that might be needed for other power purposes, such as elevators, pumps, or for heating, which are, of course, important elements in any specific problem.

The two pipe system of steam heating has long been obsolete in the west. Buildings under about eight stories high are usually supplied from basement mains, the returns coming down the supply risers. In taller buildings a steam supply line rises to the attic, which is then traversed by a supply main serving the descending risers, which carry both steam and returns. First floor radiators are sometimes supplied from the basement main, and sometimes by an independent basement main. Western achitects have been educated up to the supplying of suitable attic space for this and other necessary purposes.

The relations between the architect and engineer in the design and construction of modern buildings are worthy of more consideration than they have thus far received. Such structures are quite as much problems in engineering as they are in the architecture, and the engineer should insist upon proper recognition of his duties and responsibilities. It is to be hoped that this Society, and the others interested in the new Union Engineering building in New York, will insist upon proper recognition of the standing and dignity of the engineer. The precedent thus established will be potent and far-reaching.

Mr. George W. Colles.—The same objection lies against the titles of these papers as raised by Professor Bull against the steam-turbine papers, to wit, that they do not refer to tall office buildings in general, as they purport to do, but only to those in New York City; consequently, although much of what is said is equally true of buildings in other cities and is valuable to their designers, there is also a great deal which does not apply outside of New York. For example, the economical limit of height is stated by Mr. Bolton to be sixteen stories; this, of course, applies exclusively to New York, and then only to the down-town district and at the present time. In cities outside of New York and Chicago it is undoubtedly much less than sixteen stories, as few buildings exist of that height in other cities, and those that do exist reflect rather a penchant for advertising on the part of their owners than an attempt to get the biggest percentage of income on the investment. Although there are many buildings in New York higher than sixteen stories, it seems doubtful whether the income from the floors above sixteen stories is sufficient to pay for the cost of maintenance and fixed charges. It would have been better had we been furnished by Mr. Bolton with a regular table of cost of maintenance and fixed charges of each added

story from the bottom up, which would, of course, apply to any city; and side by side with this, the income derivable from each added story and per cent. interest on investment of the net income, which latter figures, of course, would apply only to New York, but could be recalculated for any other city, and would show us exactly where the economic limit lies in each case. But this is a question for the architect rather than the engineer.

These papers are unfortunately very fragmentary in their nature, and it seems unfortunate that the authors were not allowed more space in which to fully set forth their material on hand, as they would have been of just so much more value to those who have occasion to design plans for office buildings. Even as it is they contain about the best available information on the subject gathered together in one place, and we should be very grateful for their contribution.

Referring to foundations, I note that Mr. Wells states in this paper (paragraph 7) that the seepage amounts to less than 1,000 gallons a day. That is not extraordinary; in fact it seems quite low, but why should there be any seepage if the foundations have been well waterproofed? Nothing is stated as to the waterproofing of foundations, and those shown in the paper, Figure 6, do not show any waterproofing. I would like to ask what style of waterproofing—*i.e.*, asphalt, or tar-felt, or thick cement-mortar, or other—is used for this work, and whether any one kind has been generally adopted to the exclusion of others.

Piping should be separated into two parts, namely, the live-steam and exhaust-steam lines, as each system of piping seems to be substantially independent of the others.

I believe Mr. Wells has magnified the labors of the electrical engineer more than they deserve. There is, it seems to me, scarcely any electrical work that is more a matter of routine and which present fewer problems. There is here no question as to current systems, dynamo-types, wiring methods, etc., as these have all been standardized years ago. The direct current, two or three-wire, 110 or 220-volt system, which was the first that ever came into use commercially, has never been superseded for this class of work, and, so far as we can see, never will be. There are no problems of alternating-current or high-tension to perplex the engineer or increase the difficulty and necessity of precaution. The interior-iron-conduit system for wiring is now the only one for such buildings and probably is as good as it ever will be. The

same is true of lamps, switches, and appliances generally. In fact, the only thing in this line that the engineer needs keep up with in the way of novelty are the continual changes in the underwriters rules and city requirements, and some of the new dodges in the way of comfort and convenience.

Mr. George I. Rockwood.—Both of these papers are very much appreciated by myself. I have but two points to raise in respect to them: one, the neglect of either to mention the late and rather sudden rise in popular estimation of the plunger type of elevator for high buildings; a type which I believe is destined to supersede the electrical and the hydraulic cylinder type of machines. The motive power for providing the pressure water is of course always likely to be derived from an electrical source; but, owing to the feeling of its safety experienced by every one upon seeing the shining column under the car, connecting it at all positions with the ground beneath it, a feeling which wire cables, no matter how many are provided, can never give to quite the same satisfying extent. I believe in the plunger elevator as the final type. I might say that within the past year contracts for millions of dollars worth of plunger elevators for use in high hotel and office buildings and stores have been closed, completely filling up the manufacturing capacity of the two rival makers of these machines for years to come. I speak of these facts because I believe they are not generally known, and, as I have said, are not mentioned in either paper.

The other point to which I would call attention is the lack of ventilation in office buildings. Both owner and engineer seem to have given up the problem of efficient ventilation as insoluble, and, it would be interesting to know, why? I have no doubt the real reason is the apparent lack of general demand for and appreciation of it on the part of tenants. It is well known also to engineers that to ventilate takes three times the weight of steam required merely for heating. Reference has been made in the papers to the value for rental purposes of the floor area taken up by vertical air ducts of large size; a value which practically prohibits the use of such ducts.

The problem, however, surely must soon be solved, for there is, beneath the surface, a greater "kick' against foul-smelling offices than has yet made itself manifest to their owners. We cannot, to be sure, put down a single great fan in the sub-basement and pump air into each office by a system of ducts all con-

nected to a single large vertical riser eight or ten feet square on the ground floor; but why should not a separate complete fan system be provided on each floor? I do not see any objection to that method, and have lately adopted it in the case of a group of sixteen hospital buildings with a central station for generating electric power and steam, and in each building a hot blast heater, a supply fan and an exhauster. Why could not the same system be put up overhead?

There is one other matter—a personal issue—in connection with Mr. Wells' paper on which I should like to say a word. He refers to the so-called Van Stone pipe joint. This joint is now a standard in this country for large piping under heavy pressure; and inasmuch as I invented it and took out a patent on it, I naturally would like to see that name abandoned.

Mr. Nistle.—While I have given greater attention perhaps to the paper of Mr. Wells than to that of Mr. Bolton, fortunately for me they begin at about the same point as far as the mechanical engineer is concerned—namely, the elevator. The popular opinion that the hydraulic elevator is superior to the electric elevator for tall office buildings is voiced here, and yet I think there is good reason why we should not believe always what the people believe. If the electric elevators of to-day are not suitable in design for the attainment of the proper speed, the elevators of to-morrow or next year probably will be. I believe the real source of trouble in not getting electric elevators adapted to our use in tall office buildings is the effect on the speed produced by the methods of control that we now have. Here in Chicago we do get very good results out of the electric elevator, and we get them in all classes of buildings regardless of height or dimensions.

As to the number of elevators that we may put in an office building to get the best results out of it, I quite agree with those who look for a formula. It is not necessary to depart from what has been denominated horse-sense in this matter if we use a formula. We are not compelled to tie ourselves down to it any more than we are in the matter of steam heating. There is involved in the question not only the area, but the speed of the car; that is, its running speed, the length of time it takes to accelerate and to retard and the intervals desired between cars. Now I have developed a little formula which I have tried with pretty good success in some buildings that were erected before I came to Chicago and where weakness was found in the elevators. My

formula is simply this: basing first a calculation on the height of the building and the speed of the car, and the desired interval between cars, I make it about thus: the height in feet divided by the speed in feet per minute, multiplied by a constant which I have discovered to be about 330, divided by the desired interval in seconds between cars. Applying this, I have discovered that somebody has been trying to run a lot of good elevators on a thirty-five seconds schedule between cars, and the best he could do with the number at his disposal and the speed at which those could run varied from 40 to 50 seconds. They tried to remedy the trouble by putting in a clock, which should ring a bell for the proper time for the cars to start, but when the bell rang every 35 seconds the men went on at the old rate of 40 to 50; they could not do any better with that number of elevators and running at that speed. Then, in recognition of the very important place that area has in the calculation, I found that its variance is on the area of the cars, and I allow one square foot to about 500 square feet of the floor area. Then to get the number of cars, I take the size that I would like to make them and I put a maximum of 40 square feet on the area of one single car and a minimum of 20. Taking my first formula, based upon the height and speed, I use that for a minimum, and it gives a very small number of cars, and, in the case I have just cited, it gave a larger number than were actually installed. But by taking that as a minimum, beneath which we dare not go, and then using the area of the floor as a basis on which to build our calculations further, we can generally get just exactly the right number of cars to serve that building to the best advantage.

In the matter of the steam plant I have very little to say. We have all taken a whirl at that and have brought out some good results, and some people perhaps have brought out some poor results, but I would like to depart from the usual practice of making all units alike; I have found in the last few years that if I put in one unit of 200 kilowatts and another of 100, and still another of about 30, or possibly 50, I get just about the best service, provided always I have the assistance of suitable operators to take proper care of the plant.

H. H. Suplee.—I am very glad that Mr. Rockwood has brought up the subject of ventilation. Most of our large buildings are not prepared with any method of ventilation, but if you go into any of the tall office buildings in New York you will observe that

the tenants generally adopt some means of ventilation of their own. The only building that I know of of this class that is provided with ventilation is the Drexel Building in Philadelphia. It is a 12-story building, one of the earlier modern buildings built for business purposes, and in that building there is a complete system of exhaust ventilation. There have been built into the walls flues 3 x 12 inches made of sheet metal, with registers opening out into the rooms. These flues all run to the top of the building into mains of constantly increasing size, and these mains run from the four corners into a large exhaust fan placed in a little house on the roof, which fan is driven by an electric motor. The result is that there is a constant suction in and up to this fan, and it discharges directly upward into the open air. The rooms in the building are all heated by direct radiators placed underneath the windows.

Mr. S. H. Bunnell.—The back pressure conditions under which the steam engines of office buildings are usually operated are perhaps not sufficiently emphasized by either of the two very complete descriptions just presented. As the dark months are also the cold months, heavy lighting load and the greatest back pressure on the exhaust heating system are simultaneously required. The greatest range of expansion possible from steam at say 120 pounds is to about 3 pounds above at atmosphere, or $7\frac{1}{2}$ times. Engines for these plants are required to develop their minimum working load on this expansion ratio, and their maximum power on ratios entirely too small for compound engines. It seems to be invariably the case that the actual working loads over long periods fall much below the full load capacity of the engines; for obvious reasons where electric elevators are operated, and very considerably in the case of lighting service, since passing storms and other causes operate to suddenly vary the use of lights. Further, two or three engines are usually as many as can be provided without too much subdivision of the total capacity and too large a space requirement, so that even a steady load can not generally be carried by exactly the proper engine capacity. Sufficient stress has not been laid upon the imperative requirement of best economy over widely varying loads instead of high full load economy. Specifications call for guarantees of steam consumption at full load and without back pressure, and engine-builders can do nothing else but furnish the compound engines which show best results under the set conditions.

The non-condensing compound engine has a place wherever it can be loaded to its proper capacity; but as its minimum load and condition of best economy is with a terminal pressure near five pounds above atmospheric, thus cutting off $\frac{4}{10}$ in the high pressure cylinder, its full load capacity can hardly be double the minimum, and this with very bad efficiency. If the average load is to be about $\frac{3}{4}$ full load and not vary greatly, the non-condensing compound is all right. Under the usual conditions it would seem that simple engines are much better suited to this work, since they have a range of power from that developed at $\frac{1}{8}$ cut-off, without making the wasteful terminal loop in the indicator diagram, to that developed at say $\frac{5}{8}$ cut-off, and their maximum economy between $\frac{2}{10}$ and $\frac{3}{10}$ cut-off.

The line of improvement to follow is that of better valve action for the simple engine, and this is the idea of the four-valve single cylinder type to which brief reference has been made in the papers. It needs only a glance at the diagram of the steam distribution in a non-condensing compound engine with its gaps and other imperfections, compared with that of a four-valve simple, to suggest what is actually shown by service conditions, that compounding a non-condensing engine on a varying load is a waste of space and money.

I hope Mr. Bolton and Mr. Wells will give the engine-builders a chance to sell them simple engines which shall have a very much greater range of capacity with excellent economy, than the compound, and I am sure that such engines will come near realizing Mr. Well's computation of $2\frac{1}{4}$ cents per hour, instead of the usual rate of 4 to 10 cents.

Mr. Bryan's remarks in favor of the electric elevator are in line with my observation, that the increasing use of these machines with their varying load requirements is preventing the realization of compound-engine economy, and making the simple four-valve engine the best for the service.

Mr. R. L. Gifford.—The scope of these papers is so extended that in the short time at one's disposal it is hardly possible to refer to them even in a general way. I will simply refer to two or three matters that have been brought up in the discussion. In several places we have found it necessary to use a damp course in concrete walls, and in concrete floors in some places where the plant is placed in a sub-basement. The floor arches are reached against the thrust from below, just the reverse of ordinary floor

construction in the upper floors of a building, and in treating a concrete wall to make it impervious to seepage, if it is given two coats of hot asphaltum on the outside it will generally protect it from seepage from the outside, or even from sweating on the inside. If, on the other hand, it is desired to make a water-proof tank, the coating should be placed on the inside. It is frequently necessary to use steel plate ash pans under boilers in order to get greater head room in the ash pits under the grates, than the water level in ground would otherwise permit.

In the matter of choice between the installation of electric and hydraulic elevators, I think, it is largely a question of the service demanded by the building, and the amount of money available for the first cost of installation. Where the loads are light and variable the electric elevator, of course, is the elevator to use as the percentage of current or power required is nearly in direct proportion to the load lifted. In department stores where the load is heavy and constant, and the cars run at their full capacity all day the hydraulic elevator will give the best service with the minimum operating and repair expense. Of couse, in order to get economy in the operation of hydraulic elevators it is necessary to install high duty, and therefore very expensive pumping engines.

Respecting ventilation in office buildings, one of the great difficulties is that these buildings are sub-divided into so many small rooms that in order to get in our stacks and flues and ducts too much valuable space is used up. As a rule, in offices where there are only two or three occupants satisfactory ventilation can be obtained by simply creating an up-draft through the halls and elevator shafts, and having the transoms over the doors open and in some cases placing registers in lower panels of doors. However, in hospitals and auditoriums, theatres and banks, where the rooms are large, and, of course, in school buildings, the ventilation should always be an important part of the mechanical equipment. Hospitals, where the greatest refinement in ventilation is required, should have a system of air washing combined with the mechanical ventilation, drawing the air in from the in-take at the roof, passing it through tempering coils in the winter, then passing it through a minutely-divided water spray, after which the humidity is removed by passing it through dryers, part of the air is then run directly through the heater-coil stacks, and the balance by-passed around the heater stacks, after which it goes to

the mixing chambers, the dampers of which are controlled by thermostats in the various rooms to be heated. So that the greatest refinement can be obtained and the air delivered at any temperature that the occupants of the room may desire.

*Mr. J. H. Wells.**—It interests me very much to learn that so many engineers are interested in the mechanical problem which enters into the construction of tall buildings, and although no two of us seem to agree, still I am satisfied that no great diversity of opinion exists. The first gentleman who spoke upon this paper in criticising the theoretical layout of the plant proposes a solution which in my judgment would be wholly inadequate. In an office building in New York City of the type described, a plant such as he proposes would not furnish sufficient break-down service, and were an accident to happen to one of the one hundred kilowatt machines, I question very much whether the other two would furnish sufficient power to light and ventilate the building adequately. The demands of New York business men are so great that it would, in my judgment, be too small a plant, and competition is so strong that only the best of buildings are sought by high rent payers. The conditions in New York City are so entirely different from those in St. Louis that so light a plant as the one proposed would not be considered as a first-class office building in New York.

The criticism regarding the lack of broadness in my paper is well founded, but in the body of my paper I have stated that each particular part of the mechanical installation would require a volume if properly described.

---

* Author's closure, under the Rules.

No. 1037.*

SOME THEORETICAL AND PRACTICAL CONSIDERATIONS IN STEAM TURBINE WORK.

BY FRANCIS HODGKINSON, EAST PITTSBURG, PA.
(Member of the Society.)

1. The general subject of steam turbines is not new to the transactions of this Society—the paper presented by the late Dr. Thurston, in 1900, forms a valuable treatise on this subject. Later, at the June meting, 1903, a paper by F. A. Waldron, was presented, which, from the operator's standpoint, showed the all-around commercial economy of the turbine, and was a tribute to the durability and general performance of the particular machine he described. It is therefore intended to confine this paper to some special features of turbine work, which have a more or less intimate bearing upon the operation of a successful steam turbine.

The Ideal Turbine Element.

2. An ideal elementary turbine may be said to consist of a steam nozzle directing a tangential jet of fluid upon a bucket wheel of the Pelton type. This type of bucket is selected because it may be capable of giving a complete reversal to the jet, so that the spent fluid may issue from the buckets without any velocity. Even this type may not be quite ideal because of the distance which must necessarily exist between the outlet of the nozzle and the receiving wedge of the buckets, the effect of which will be to cause friction between the jet and the surrounding medium, caus-

---

* Presented at the Chicago meeting (May and June, 1904) of the American Society of Mechanical Engineers, and forming part of Volume XXV. of the *Transactions*.

† For further discussion on this topic, consult *Transactions* as follows:
No. 345, vol. x., p. 680 : "Notes on Steam Turbine." J. B. Webb.
No. 648, vol. xvii., p. 81 : "Steam Turbine." W. F. M. Goss.
No. 876, vol. xxii., p. 170 : "Steam Turbine." R. H. Thurston.
No. 987, vol. xxiv., p. 999 : "Steam Turbine from Operating Standpoint." F. A. Waldron.

TESTS OF WESTINGHOUSE-PARS

SIZE OF TURBINE																					
VACUUM IN EXHAUST	26 Inches.										27 Inches.										
THROTTLE PRESSURE LBS. GAUGE	125			140				150			150										
CONDITION OF THE STEAM	Dry and Saturated.			Quality = 99%.				Dry and Saturated.			Dry and Saturated.								5° Superheat		
REVOLUTIONS PER MINUTE	3,600			3,600				3,600			3,600								3,600		
TURBINE NUMBER	28			35				51			47			49			55		53		
1 Steam pressure lbs. per gauge	125.6	124.9	130.6	142.8	144.0	144.5	138.2	151.6	151.4	149.8	153.4	153.2	153.1	152	156	155	* 152	* 155	140.9	148.3	154
2 Vacuum referred to 30-inch barometer	26.0	26.02	26.03	26.0	26.0	26.0	26.0	26.06	26.06	26.06	27.04	27.11	27.06	27.15	27.2	27.25	27.01	27.01	27.0	27.0	27
3 Superheat, degrees Fahrenheit	2.5	...	...	...	...	...	...	...	2.0	4.0	...	...	...	3.0	5.0	...	...	...	7.0	5.0	4
4 Quality of the steam	...	.999	.999	.989	.989	.99	.948	1.00	...	...	1.00	1.00	.998	...	...	.997	1.00	1.00	...	...	...
5 Revolutions per minute	3,536	3,555	3,545	3,511	3,557	3,582	3,589	3,541	3,599	3,593	3,528	3,576	3,583	3,520	3,565	3,577	3,497	3,549	3,541	3,563	3,564
6 Load in kilo-watts	...	...	...	...	...	...	...	...	...	...	438.1	298.4	211.6	...	...	...	...	...	...	...	...
7 Load in electrical horse-power	...	...	...	...	...	...	...	...	...	...	587.3	400.0	283.6	...	...	...	...	...	...	...	...
8 Load in brake horse-power	580.1	457.2	326.0	623.4	508.2	381.0	10.7	610.8	451.4	266.1	...	...	...	651	435	261	691	593.5	607.7	441.2	288.9
9 Total pounds of steam per hour	8,940	7,481	5,831	9,502	7,992	6,311	1,035	8,917	6,916	4,542	8,739	6,205	4,774	9,200	6,472	4,411	9,734	8,514	8,882	6,625	4,774
10 Pounds of steam per E. H. P. hour	...	...	...	...	...	...	...	...	...	...	14.89	15.51	16.81	...	...	...	...	...	...	...	...
11 Pounds of steam per B. H. P. hour	15.41	16.36	17.89	15.24	15.73	16.56	...	14.59	15.32	17.06	...	...	...	14.15	14.88	16.9	14.1	14.35	14.61	15.01	16.52
12 Load in per cent. of full load	98	77	55	105	86	64	2	103	77	45	109	75	53	110	74	44	119	100	103	75	49
	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21

SIZE OF TURBINE	400 K. W.												1,000 K. W.						
VACUUM IN EXHAUST	28 Inches.												27 Inches.				25 Inches.		
THROTTLE PRESSURE LBS. GAUGE	150												130				150		
CONDITION OF THE STEAM	100° Superheat				150° Superheat.			180° Superh		190° Superheat.			Quality = 100%				Quality = 98%.		
REVOLUTIONS PER MINUTE	3,600				3,600			3,600		3,600			1,500				1,500		
TURBINE NUMBER	55				54			55		67			15				17		
1 Steam pressure lbs. per gauge	* 150	* 150.3	* 154.5	* 153.3	150.6	149.4	150.2	* 150.2	* 153.6	148.4	147.9	149.1	132.5	132.7	134.3	139	148.7	150.5	151
2 Vacuum referred to 30-inch barometer	28.01	28.01	28.01	28.01	28.0	28.0	28.0	28.08	28.1	28.0	28.0	28.0	27.02	27.0	27.08	26.83	25.07	25.08	25.18
3 Superheat, degrees Fahrenheit	104	104	106	88	139	158.8	150	182	180	192	185	157	...	...	...	...	...	...	...
4 Quality of the steam	...	...	...	...	...	...	...	...	...	...	...	...	.999	.999	.999	.998	.98	.98	.98
5 Revolutions per minute	3,458	3,544	3,581	3,627	3,538	3,586	3,595	3,478	3,543	3,551	3,571	3,583	1,489	1,498	1,508	1,513	1,498	1,503	1,500
6 Load in kilo-watts	...	...	...	...	...	...	...	...	...	428.4	303.8	197.8	1,101.7	786.7	530.7	380.1	1,506	914	310
7 Load in electrical horse power	...	...	...	...	...	...	...	...	...	571.5	407.2	265.1	1,476.8	1,054.5	711.4	509.4	2,015	1,224	417
8 Load in brake horse-power	759	593	445	264	626	451	296.8	763	592	...	...	...	...	...	...	...	...	...	...
9 Total pounds of steam per hour	9,157	7,408	5,728	3,504	7,426	5,324	3,919	8,520	6,795	7,444	5,626	4,126	21,984	16,867	12,775	10,547	32,924	23,030	11,713
10 Pounds of steam per E. H. P. hour	...	...	...	...	...	...	...	...	...	13.02	13.81	15.56	14.9	15.99	17.95	20.75	16.32	18.82	28.1
11 Pounds of steam per B. H. P. hour	12.06	12.5	12.87	14.48	11.86	11.87	13.2	11.17	11.46	...	...	...	...	...	...	...	...	...	...
12 Load in per cent. of full load	128	100	75	45	106	76	50	129	100	106	76	50	110	79	53	38	120	73	25
	44	45	46	47	48	49	50	51	52	53	54	55	56	57	58	59	60	61	62

SIZE OF TURBINE	1,250 K. W.																
VACUUM IN EXHAUST	28 Inches.			25 Inches.		26 Inches.		27 Inches.									
THROTTLE PRESSURE LBS. GAUGE	150			150		150		150									
CONDITION OF STEAM	140° Superheat.			Dry Saturated.		Dry Satur'd.		Dry Saturated.									
REVOLUTIONS PER MINUTE	1,500			1,200		1,200		1,200								1,200	
TURBINE NUMBER	17			43		43		41								43	
1 Steam pressure lbs. per gauge	154	149.5	148.9	146.4	148.4	147	148.9	‡ 141.8	‡ 144.5	‡ 148	‡ 147.1	‡ 146.5	‡ 146.8	‡ 151.4	‡ 150.3	146.3	145.9
2 Vacuum referred to 30-inch barometer	28.1	28.04	28.17	25.01	25.05	26.01	26.05	26.79	26.05	27.05	27.11	27.11	27.11	27.1	27.08	27	27
3 Superheat, degrees Fahrenheit	141	139	138	2	3	1	...	...	...	...	...	...	...	...	...	2	2
4 Quality of the steam	...	...	...	...	...	...	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	.999	...	...
5 Revolutions per minute	1,500	1,500	1,503	1,199	1,201	1,198	1,202	1,197.4	1,200	1,201	1,196	1,197	1,199	1,201	1,201	1,199	1,199
6 Load in kilo-watts	1,512	904	306	...	...	...	...	1,988.9	1,713.5	1,489.4	1,321.5	989.5	656	342.7	197	...	...
7 Load in electrical horse-power	2,030	1,210	412	...	...	...	...	2,666	2,297.1	1,996.5	1,771.4	1,326.5	879.3	459.4	264	...	...
8 Load in brake horse-power	...	...	...	2,008.3	1,004.4	2,002.3	1,007	...	...	...	...	...	...	...	...	2,005.3	1,499.7
9 Total pounds of steam per hour	25,685	17,042	8,108	28,902	16,809	28,236	16,106	40,547	33,036	28,208	25,712	20,254	15,074	9,732	7,150	27,248	21,189
10 Pounds of steam per E. H. P. hour	12.66	14.1	19.68	...	...	...	...	15.21	14.38	14.13	14.52	15.27	17.14	21.18	27.08	...	...
11 Pounds of steam per B. H. P. hour	...	...	...	14.39	16.73	14.10	15.99	...	...	...	...	...	...	...	...	13.59	14.13
12 Load in per cent. of full load	121	72	24	110	55	110	55	159	137	119	106	79	53	28	16	110	82
	86	87	88	89	90	91	92	93	94	95	96	97	98	99	100	101	102

* Test made by Mr. F. W. Dean of Dean & Main, Boston. † Tests made by a board of naval engineers. ‡ Tests wi

400 K. W.																						
...nches.									28 Inches.													
150									150													
...uperheat.			100° Superheat.						Dry Saturated.								45° Superheat.					
3,600			3,600						3,600								3,600					
53			37			39			55					30			52					
154	153.5	154	119.8	160.6	152.8	155.2	147.5	155	*154.6	*154.5	*156	*155.8	*152	151.7	154.5	153	143.5	149.6	150.3	152.1	150.7	150
27	27.0	27	27.01	27.0	27.0	27.05	27.0	27	28.01	27.98	27.92	28.01	28.01	28.0	28.0	28	27.78	28.03	28.0	28.01	28.02	28.02
4	2.0	.....	100	103.5	95.6	97.6	104.2	103.1	.....	.....	.....	.....	.....	.....	.....	.....	45.0	44.0	42.0	45.0	41.0	45.0
.....	.....	.998	.....	.....	.....	.....	.....	.....	1.00	.999	1.00	.996	.986	.992	.992	1.00	.....	.....	.....	.....	.....	.....
3,564	3,583	3,602	3,492	3,570	3,606	3,482	3,509	3,530	3,480	3,545	3,583	3,602	3,662	3,528	3,570	3,591	3,414	3,436	3,478	3,511	3,526	3,549
.....	.....	.....	.....	.....	.....	411.2	311.7	219	.....	.....	.....	.....	.....	402	292.2	198.7	.....	.....	.....	.....	.....	.....
.....	.....	.....	.....	.....	.....	551.1	417.9	293.7	.....	.....	.....	.....	.....	539	391.7	266.3	.....	.....	.....	.....	.....	.....
288.9	152.6	9.2	650.3	481	254.4	.....	.....	.....	729	593.2	448	241	5.3	.....	.....	.....	1,028.2	824.6	755.9	621.5	424.6	258.1
4,774	2,881	953.5	8,778	6,657	4,039.5	7,425	5,780	4,272	9,928	8,269	6,488	3,876	706	8,169	6,172	4,640	14,537	11,020	9,998	8,417	6,015	4,015
.....	.....	.....	.....	.....	.....	13.47	13.83	14.54	.....	.....	.....	.....	.....	15.15	15.75	17.42	.....	.....	.....	.....	.....	.....
16.52	18.88	.....	13.5	13.84	15.87	.....	.....	.....	13.62	13.91	14.48	16.05	.....	.....	.....	.....	14.13	13.36	13.23	13.56	14.16	15.55
49	26	.....	110	82	43	103	78	55	123	100	76	41	1	100	73	50	174	140	129	105	72	44
21	22	23	24	25	26	27	28	29	30	31	32	33	34	35	36	37	38	39	40	41	42	43

1,250 K. W.																							
...es.	26 Inches.			27 Inches.																	28 Inches.		
	150			150													150				150		
...98%.	Quality = 98%.			Quality = 98%.			Quality = 99%.				50° Superheat.				Dry Steam.		25° Superheat.				Quality = 98%.		
	1,500			1,500			1,200				1,500				1,500		1,500				1,500		
	17			17			32				17				31		31				17		
151	150.5	150	151	147	150.4	148	147.7	146.5	148.6	149.2	149.8	149.6	149.5	151	†149.4	†146.2	146	†144.3	147.6	155.1	148.8	150	151.8
25.18	26.05	25.92	26.22	26.93	26.99	26.87	27.05	27.06	27.07	27.08	27.01	26.92	27.06	26.68	27.5	27.85	27.48	27.52	27.46	27.55	27.42	28.12	28.98
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	50.0	51.0	49.0	54.0	.....	.....	28	28	28	30	.....	.....	.....
.98	.98	.98	.98	.98	.98	.98	.992	.994	.992	.991	.....	.....	.....	.....	.994	.995	.....	.....	.....	.....	.988	.975	.979
1,500	1,503	1,503	1,500	1,495	1,502	1,500	1,186	1,203	1,206	1,221	1,502	1,503	1,500	1,500	1,456	1,477	1,476	1,491	1,503	1,507	1,499	1,502	1,500
310	1,519	897	306	1,529	963	254	1,530.2	1,154.5	792.1	392.5	1,475	1,253	877	302	1,510.4	1,019.1	1,557.6	1,239.8	767.9	383.4	1,545	935	262
417	2,038	1,201	410	2,050	1,285	340	2,051.2	1,547.8	1,061.6	526.1	1,978	1,681	1,173	405	2,024.6	1,365	2,087.8	1,662	1,028.1	513.9	2,065	1,252	249.2
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
11,713	32,039	21,985	10,891	31,192	21,918	9,109	30,358	23,233	18,273	10,625	27,246	24,544	19,223	9,773	28,333	20,915	28,537	23,181	16,507	10,302	3,456	19,703	7,392
28.1	15.72	18.29	26.62	15.22	17.06	26.79	14.8	15.01	17.21	20.19	13.82	14.6	16.35	24.2	13.99	15.31	13.67	13.94	16.05	20.04	14.73	15.75	21.8
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
25	121	72	24	122	77	20	123	92	63	31	118	100	70	24	120	81	125	99	61	31	124	75	21
62	63	64	65	66	67	68	69	70	71	72	73	74	75	76	77	78	79	80	81	82	83	84	85

1,250 K. W.																				2,600 K. W. §		
							28 Inches.													27.5 Inches.		
							150													190		
		75° Superheat.					Dry Saturated.					75° Superheat.				100° Superheat.				190° Superheat.		
,200		1,200					1,200					1,200				1,200				1,360		
43		41					41			43		41				43						
145.9	148.7	‡146.3	‡147.7	‡151	‡151.9	‡151.8	‡146.1	‡146.8	‡151	146.9	149	‡146	‡147.6	‡150.8	‡151.8	147	146.4	147.2	148.8	188	188	186
27	26.95	27.1	27.1	27.15	27.07	27.15	28.08	28.1	28.05	28.1	28	28.1	28.1	28.05	28.05	27.95	28.02	27.97	28.02	27	27.5	27.98
2	2	76	76	77	77	77	.....	.....	.....	2	2	78	77	77	76	102.3	104	102.3	102	214	180	186
.....	.....	.....	.....	.....	.....	.....	1.00	1.00	1.00	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
1,199	1,202	1,201	1,205	1,209	1,213	1,201	1,197	1,201	1,198	1,200	1,204	1,199	1,204	1,214	1,217	1,203	1,202	1,203	1,199	1,360	1,360	1,360
.....	.....	1,293.9	986.2	664.7	333.6	191	1,364	972	334.8	.....	.....	1,274.2	977.6	333.2	198.4	.....	.....	.....	.....	2,995	2,518	1,945
.....	.....	1,734.4	1,322	891	447.1	256	1,828.3	1,303	448.7	.....	.....	1,708	1,310.5	446.6	266.1	.....	.....	.....	.....	4,010	3,378	2,608
1,499.7	1,004	.....	.....	.....	.....	.....	.....	.....	.....	1,578.2	911.5	.....	.....	.....	.....	1,808	1,655.8	1,298.4	815.4	.....	.....	.....
21,189	15,241	23,903	19,108	14,181	9,170	6,734	25,639	19,334	9,295	21,286	13,337	22,504	18,180	8,420	6,300	23,180	21,302	17,009	11,671	44,236	39,356	30,867
.....	.....	13.78	14.45	15.9	20.51	26.3	14.02	14.91	20.71	.....	.....	13.17	13.87	18.86	23.68	.....	.....	.....	.....	11.02	11.57	11.83
14.13	15.18	.....	.....	.....	.....	.....	.....	.....	.....	13.49	14.63	.....	.....	.....	.....	12.82	12.86	13.09	13.70	.....	.....	.....
82	55	104	79	53	27	15	109	78	27	87	50	102	78	27	16	99	90	71	34	115	97	75
102	103	104	105	106	107	108	109	110	111	112	113	114	115	116	117	118	119	120	121	122	123	124

...ests witnessed and verified by engineers of the staff of Julian Kennedy, Pittsburg, Pa.

§ Built by the Brown-Boveri Company.

ing eddies, and perhaps entraining some of this surrounding medium.

3. Steam expands approximately adiabatically in the nozzle, and on its arrival at the exit should be thoroughly expanded to the exhaust pressure, so that its heat energy may be entirely transformed into kinetic energy, and its velocity be the greatest attainable on striking the buckets. For conditions of maximum efficiency, the same relation between velocities of jet and vane applies to the steam impulse wheel, as to its hydraulic analogue; viz., bucket speed equals one-half of jet speed.

4. The difficulty of such a design as this is obviously due to the fact that the velocity of steam expanding between even moderate differences of pressure, is so high as to render the most efficient velocity of the bucket difficult to provide for because of the limitations of strength of materials. Thus, for terminal pressure of 165 pounds and 1 pound respectively, the bucket speed should be 2.025 feet * per second. In practice, the maximum attained is 1,378 feet per second in a 300 horse-power De Laval turbine, which is 32 per cent. below speed. In a 50 horse-power turbine, the bucket velocity is 58 per cent. below normal, but even at this speed, the radial stresses amount to 23,000 pounds per square inch.

The Expansion of Steam.

5. The so-called phenomenon of maximum flow of steam through an orifice is now generally understood. It has been generally stated that the velocity of steam flowing through an orifice will not exceed about 1,500 feet per second, no matter what the differences of pressure may be. This condition has been said to be reached when the exhaust pressure bears a certain ratio to the initial pressure. This ratio has been variously given from 52 per cent. to 58 per cent.

6. Experiments made at East Pittsburg, however, seem to show that the above figures vary with the initial pressure. The velocity becomes higher and the above ratio lower with the lower initial pressure. Therefore, in the above-mentioned elementary turbine, if the initial pressure is 150 pounds per square inch, and the exhaust pressure is greater than 85 pounds (57 per cent. of the initial pressure), all that is needed in the shape of a nozzle is a well-rounded orifice. If the exhaust pressure is less than 85

* Conrad Anderson—Trans. Engrs. & Shipbuilders, Scotland, November, 1902.

pounds, a divergent nozzle, as used by Gustav De Laval, is necessary. Without this divergence, the steam will expand outside of the nozzle where it is uncontrolled by its walls, and consequently much of the expansion will take place laterally, which will not further accelerate the jet.

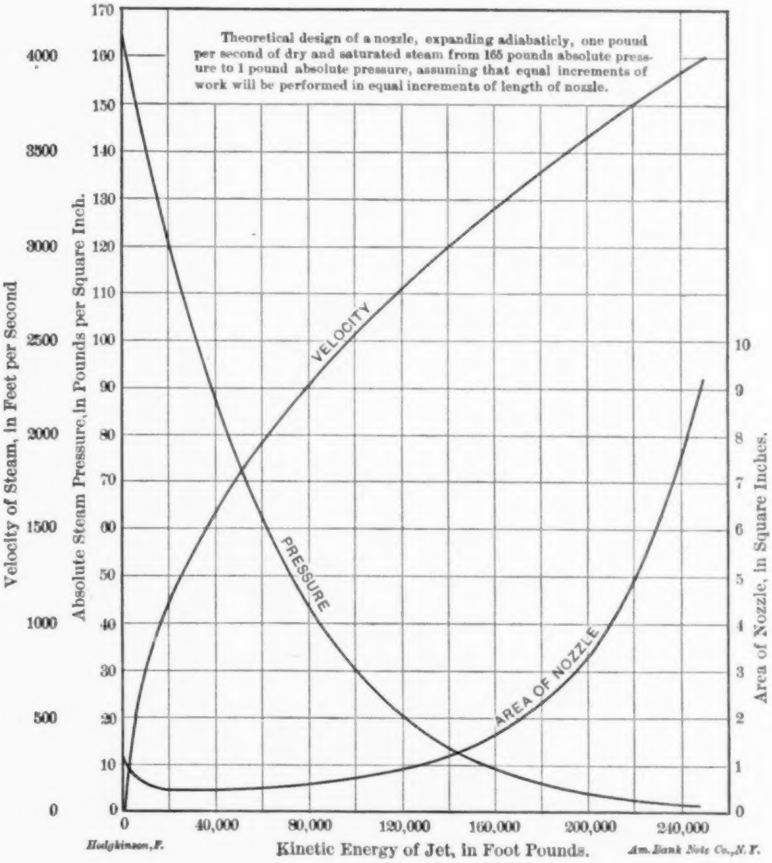


FIG. 357.

7. A further reason for the divergence is apparent from the fact that the work done by the expanding steam varies nearly in direct proportion to the number of expansions, and also varies directly as the square of the velocity. The volume of the steam, therefore, increases much more rapidly than the velocity, and room must be provided for its proper expansion in the nozzle.

8. A theoretical design of a divergent nozzle is shown in Fig. 357, which considers a nozzle expanding 1 pound of dry saturated steam per second from an absolute pressure of 165 pounds per square inch to 1 pound per square inch. It has been assumed that the expansion will be adiabatic, and that equal increments of work will be performed in equal increments of nozzle length.

9. The abscissae are laid out to uniform scale and represent the foot pounds of energy given up by the steam expanding between the limits of pressure referred to the scales of ordinates. The curve of pressure shows the pressure drop corresponding to increasing work. The velocity curve is calculated upon the assumption that all energy of the expanding steam has been expended internally and converted into velocity. The curve of the areas of the nozzle is found by the equation $a = \frac{s \times x_2}{V}$.

$$
\begin{aligned}
a &= \text{Area of nozzle,} \\
s &= \text{Specific volume,} \\
x_2 &= \text{Quality of steam,} \\
V &= \text{Velocity of steam.}
\end{aligned}
$$

10. An examination of the curve shows that the throat of the nozzle corresponds to a velocity of 1,500 feet per second and to a pressure $57\frac{1}{2}$ per cent. of the initial pressure.

11. It is logical to assume that steam expands adiabatically in a nozzle, as nozzles are generally small relative to the amount of steam passing, so that there is little opportunity for any interchange of heat. The state of the steam at the outlet of the nozzle, therefore, is, except for some frictional losses, the state of the exhaust of an ideal engine. It contains considerable water which has been condensed during the adiabatic expansion of the steam. This may amount to over 20 per cent. in cases of low exhaust pressures, and is discharged from the nozzle in the form of vapor or fine spray.

12. If the steam issuing from this nozzle be brought back to rest in a closed chamber, the kinetic energy of the jet will be reconverted into heat, and the expanded steam will become superheated, just as in the well-known throttling calorimeter. Fig. 358 shows this graphically, by means of an entropy temperature diagram. The steam expands from 165 pounds absolute to 1 pound absolute. S represents the state of the steam at the outlet of the nozzle, which contains 23.1 per cent. of moisture. On the

steam being brought to rest, it attains the state S_2 and becomes superheated 167.8 degrees Fahr.

13. An interesting fact on this subject was pointed out by Prof. W. H. Watkinson (Transactions of the Institute of Engineers and

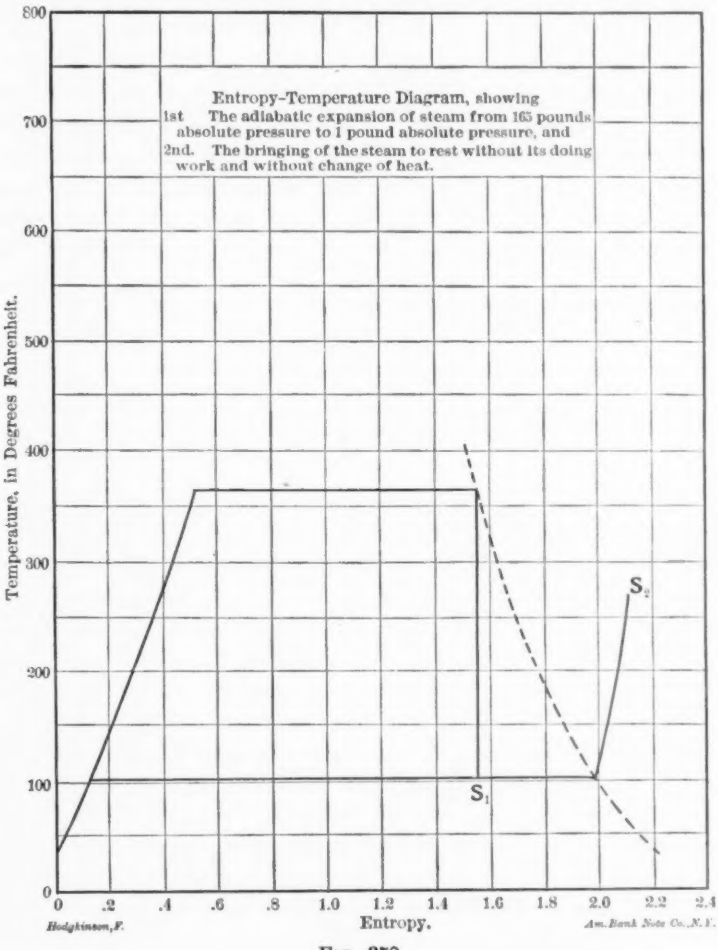


FIG. 358.

Shipbuilders in Scotland, vol. xlvi., Part V.), of England, who shows that if, after the energy of the jet has been converted into velocity, we were to separate therefrom the water of condensation and then bring the jet to rest, the steam would become highly superheated. This is graphically shown in Fig. 359. Using the

same limits of pressure as heretofore, the steam on being brought to rest rises in temperature 670.1 degrees Fahr., and attains 772.1 degrees Fahr., which is 406.2 degrees Fahr. hotter than when it left the boiler.

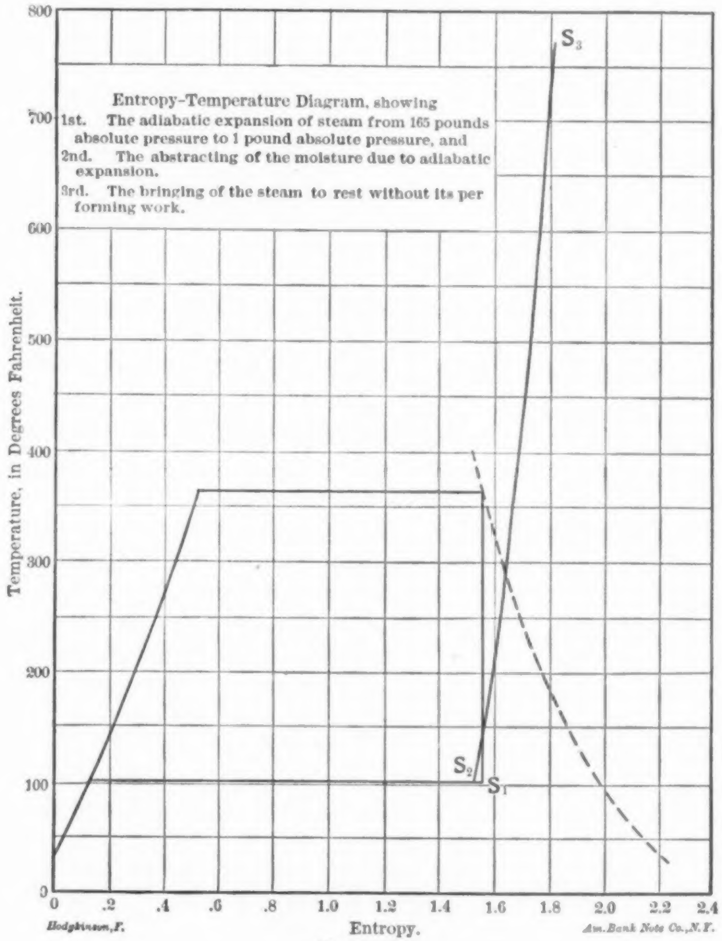


Fig. 359.

14. This method might be utilized for obtaining superheated steam at a moderate pressure, if any practicable method could be devised for separating the water of condensation from the jet before it is brought to rest. For instance, operate the boiler at 400 or 500 pounds pressure and expand in the manner above outlined

to 150 pounds per square inch, then separating the moisture and bringing the steam to rest, superheat would be obtained. An analysis of the cycle will show this method to have about the same efficiency as though the steam were generated in the boiler at 150 pounds and afterwards superheated to a corresponding degree.

Causes of Departure from the Ideal.

15. It is probable that the state of steam issuing from a good-sized nozzle does not deviate much from the ideal. The only losses will be due to the friction and conduction of heat along the walls. The larger the nozzle, the less will be the ratio of perimeter to cross section, and consequently the less these losses will become. In the ideal turbine above mentioned, if the form and the velocity of the buckets were such that they might wholly absorb the velocity of the jet without friction, the state of the steam on leaving the buckets would be the same as when it left the nozzle, and we would have a turbine approximating an ideal engine, except for the frictional losses of the nozzle.

16. In practice, the kinetic energy cannot be wholly absorbed, the buckets cannot generally be arranged to satisfactorily make a complete reversal without attendant disadvantages; hence the steam issues from the bucket with residual velocity. Where the jet is arranged to impinge upon the bucket wheel tangentially or at the sides, a departure from the ideal occurs. In the De Laval type, complete reversal cannot be obtained, as the spent steam must clear the buckets; in the Pelton type, the angular position of the bucket with reference to the jet is continually changing, resulting in distortion of the jet from its ideal path. There is also in some forms a spill from the buckets, and always frictional losses and eddies, which have the effect of lowering the velocity and heating up the steam similar to, but to a less extent than, the effect in a throttling calorimeter.

17. With high steam velocities, the skin friction of the fluid passing over surfaces such as buckets amounts to a considerable loss, though the exact amount of such losses is unknown.

Professor Perry says: *

"Friction in fluids is proportional to the speed when the speed is small; to the square of the speed when the speed is greater; and at still greater speeds, the friction increases more rapidly than the square of the speed.

The resistance to motion of a rifle bullet is proportional to the square root of the fifth power of the speed."

* See "Applied Mechanics," by Prof. John Perry, p. 79.

In the type of turbine above described, the velocity of the steam is approximately twice that of a rifle bullet.

18. Dr. Stodola, in his excellent work, "Die Damp Turbine," quotes a test by Lewici upon the friction of a 30 horse-power De Laval disk in air and vacuum. The turbine was driven by a calibrated motor, and in order to bring it up to speed, the following power was required:

		Resistance.	
		Atmosphere.	19.6" Vacuum.
Dry saturated steam......	3.3 H.P. =	11 per cent.	5 per cent.
Superheat 300 degrees C..		6.25 per cent.	2 per cent.

19. In view of these facts, it would seem desirable to avoid high steam velocities as much as possible, because of the resulting frictional losses. Another reason is the erosive action of the steam with high velocities. This is quite serious when the steam is initially wet, due to foaming boilers. In this case, matters are generally made worse by the moisture generally carrying with it various kinds of solid impurities.

20. In this connection, the author lately had some hard drawn Delta metal blades exposed to two steam jets, the one issued from a diverging nozzle with 150 pounds boiler pressure behind it, and the other from a rounded orifice with 1 pound pressure. The size of the outlet of the two nozzles is the same in each case, and the respective velocities were approximately 2,900 and 600 feet per second.

21. The blades were kept continuously exposed to the jets for one hundred and twenty-eight hours.

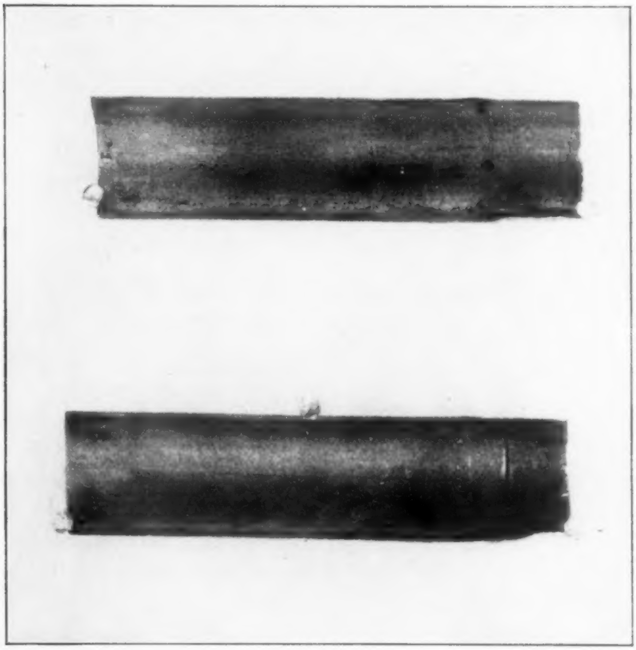
22. Fig. 360 is a photograph of their condition, in which some considerable amount of erosion will be observed on the blades subjected to the higher velocity.

23. A rather curious feature, too, is that the erosion was maximum at the center and the extreme edges of the jet.

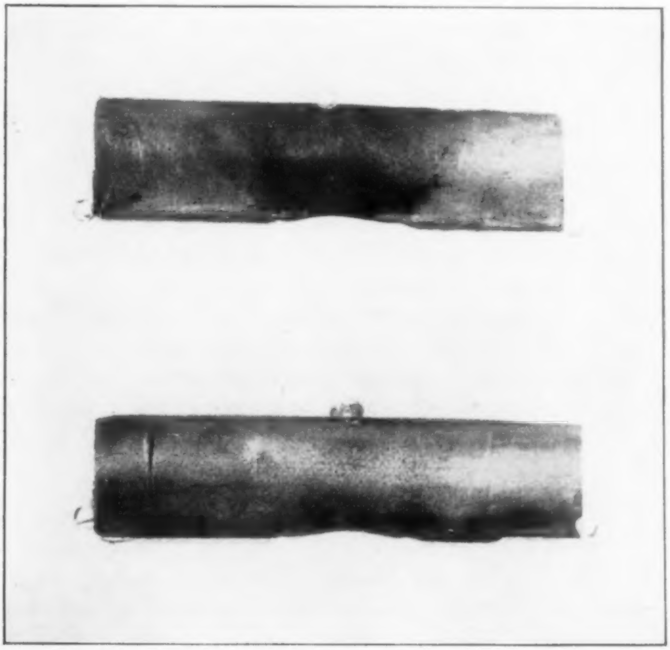
24. No attempt was made to observe the quality of the steam; the nozzles were merely connected to a steam pipe in the works.

Types of Turbines.

25. The ideal turbine that has been above referred to is essentially a refined form of that built by Branca, in 1629. The present De Laval turbine may be said to be of a like type, in so far that its elements consist of one single bucket wheel and one set of expanding nozzles.



Subjected for 128 hours to 600 feet per second velocity steam.



Subjected for 128 hours to 2,900 feet per second velocity steam.

FIG. 360.—PHOTOGRAPHS OF JET TURBINE BLADES SHOWING EROSION.

The Stumpf:

26. A more recently developed turbine, Stumpf, likewise utilizes these distinguishing elements of the simple impulse turbine, with, however, a slight departure from the De Laval arrangement in the use of tangential nozzles with buckets of the Pelton form milled in the periphery of the wheel. These are slightly pitched from the shaft in order to provide metal for succeeding buckets. A characteristic of this form of turbine is the large wheel diameter employed, and it is in this direction that low shaft speeds are secured. For example, a 500 horse-power turbine with two disks 5 feet in diameter has been constructed and run at a speed of 3,000 revolutions per minute with a rigid shaft. In later forms of the Stumpf turbine, it has been sought to utilize a residual velocity of the steam leaving the buckets by redirecting it around a circular guide, so as to impinge again upon another set of buckets, milled in the periphery of the wheel, alongside the other row. The steam jet, after leaving the nozzle, is, therefore, reversed 180 degrees twice before finally leaving the wheel. It is evident, however, that this arrangement is subject to the same disadvantage as regards fluid friction, as the simple impact arrangement.

Compound Systems:

27. On account of the difficulties of construction, and losses attendant upon the use of high steam velocities, various methods of compounding have been proposed, the objective point of the compounding arrangement being a subdivision of total velocity among several stages, so that the working velocities in any one stage might be reduced to a more practicable degree. The effect of compounding does not, however, reduce the stage velocities as promptly as might be at first thought, for the reason that the velocity varies as the square root of the energy of the steam. Thus, if the simple impact turbine were constructed with two stages instead of one, the stage velocity would be reduced from 4,012 feet per second to 2,835 feet per second, assuming a range of pressure from 165 pounds to 1 pound absolute. If it were constructed in four stages, the velocity would become 2,050 feet per second, under the supposition that the entire velocity of the jet were abstracted in each stage. In order, therefore, to reduce the steam velocities to 500 feet per second, about 64 stages would be required in the turbine.

The Curtis:

28. The nearest approach to this method of compounding the simple impact element is carried out in the Curtis turbine, in which two or more stages are employed to carry out the total range of expansion from boiler to condenser. Each stage comprises a set of expanding nozzles and a wheel carrying more than one row of buckets. The peripheral speed of the wheel is kept within convenient limits, so that if only one row of buckets were employed, the steam mould issue from it with much residual velocity.

29. In order to abstract this as far as possible, a set of guides

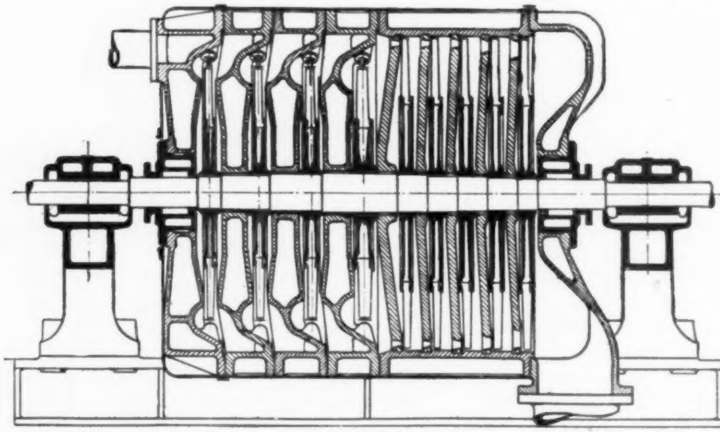


FIG. 361.

is interposed which redirects the steam leaving the first row of moving blades, into a second set. The velocity of the jet is thus reduced by each reversal in the moving blades, and this process is carried out as many times as may be necessary to absorb the initial velocity of the jet. Thus the steam in each stage is alternately accelerated in the nozzle and retarded in the blades. Low shaft velocities are secured by this arrangement, but this is largely due, however, to the use of large diameters of wheel, and to the fact that, generally, a very small arc of all but the last wheel is being acted upon by the nozzles at one time. This construction makes desirable a fine axial clearance.

The Zoelly:

30. The idea of compounding has been applied in a slightly different manner in the Zoelly turbine, a cross section of which is

shown in Fig. 361. One section of the turbine—the high-pressure end—consists of several tangential impulse elements arranged in separate compartments. Each is fitted with a number of nozzles in which part of the expansion is carried out, and the velocity immediately abstracted in the buckets. In the later stages, however, a different construction is employed by reason of the increased volume of steam. The nozzles are here slotted in the wall of each compartment, and the steam flow is in the form of an annular jet, striking the bucket wheel in the manner of the De Laval turbine. To accommodate expansion, the radial widths of the nozzle parts are progressively increased to the end of the turbine. The principal features of the Zoelly turbine lies, how-

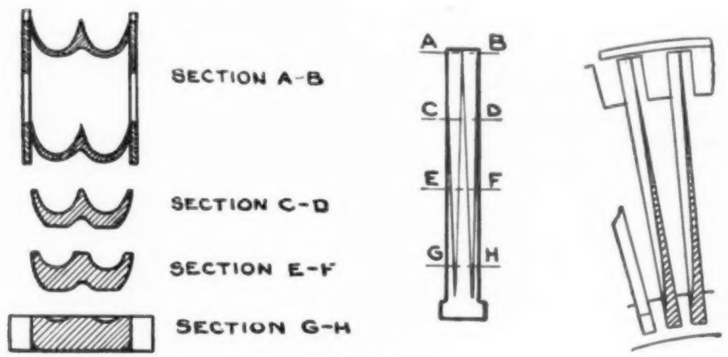


Fig. 362.

ever, in the construction of the disk wheel to accommodate extraordinarily high peripheral speeds. The buckets consist of metal strips about one-half the radius of the wheel length. They are secured to a two-piece hub by projections, as shown in Fig. 362, and a cross section of approximately uniform strength is obtained by milling the bucket curves deeper from hub to rim. Thus, the weight is largely decreased, as are also the internal stresses to be provided for. Zoelly found by experiments that his buckets might be spaced much farther apart than customary in the De Laval turbine, without serious loss in efficiency, which justifies his construction. The peripheries of the bucket wheels are surrounded by stationary metal shrouds to prevent radial escape of the steam. The sides of the buckets are further enclosed in sheet-steel housings to reduce the windings which would occur with exposed buckets of such length. Complete reversal of

the steam jet is impossible, and the action is similar to that of the familiar Pelton wheel.

The Rateau:

31. Professor Rateau, in his later form of turbine, has discarded the simple impact element and employed the subdivided element to a still greater extent than any hereinbefore described. Fig. 363 shows a section of a Rateau turbine of 25 stages. Annular nozzles are provided in each division wall between stages. The stages being numerous, the pressure drops are small, such that the nozzles require no divergence. The increasing nozzle area is secured through increasing the arc, or percentage of total cir-

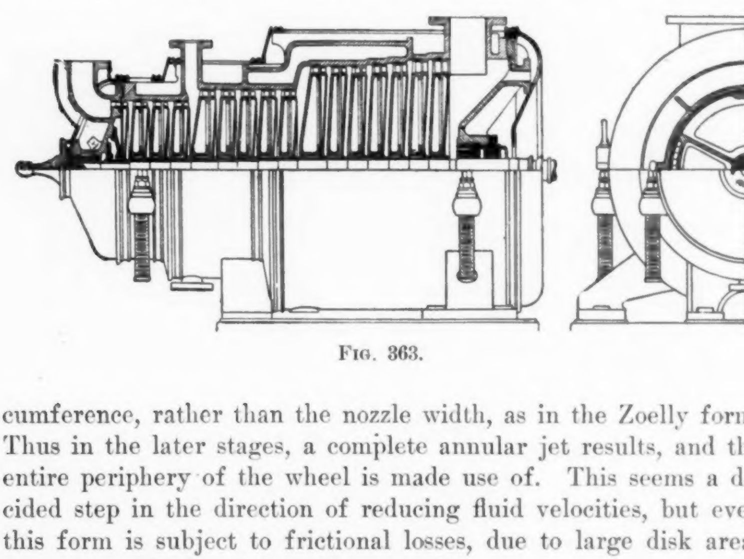


Fig. 363.

cumference, rather than the nozzle width, as in the Zoelly form. Thus in the later stages, a complete annular jet results, and the entire periphery of the wheel is made use of. This seems a decided step in the direction of reducing fluid velocities, but even this form is subject to frictional losses, due to large disk areas operating at high speeds in dense media.

32. A still further method of securing the advantages of compounding is represented in the construction of the Parsons turbine, which, however, antedates, both in conception and introduction, other forms of modern steam turbines.

Parsons:

33. In the types previously described, the original impact element has been made use of in simple or compounded form, the pressure fall being secured by nozzles and the velocity abstracted by vanes. Thus each bucket wheel presumably rotates in an at-

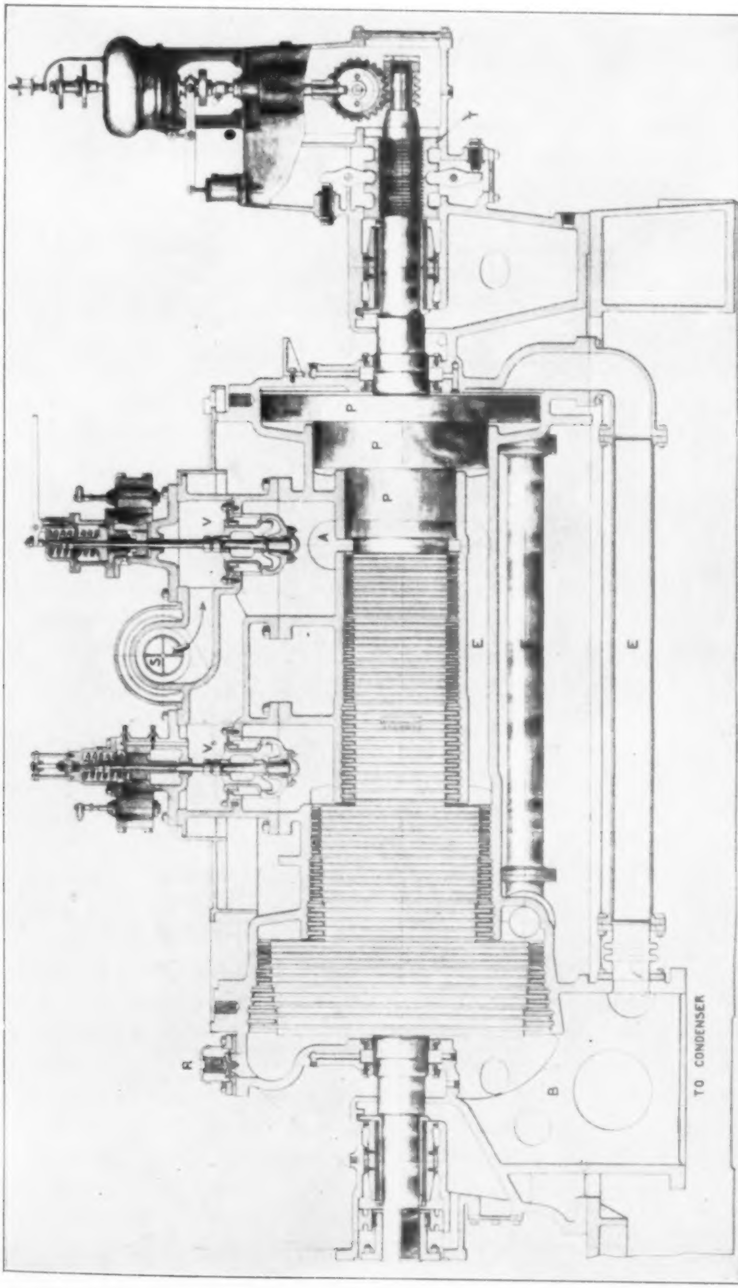


Fig. 364.—Typical Westinghouse-Parsons Steam Turbine.

mosphere of uniform pressure at all points. Mr. Parsons, however, early conceived the idea of so designing and locating the turbine vanes that they should perform the functions of bucket and nozzle as well, at the same time confining the steam to the periphery of the wheel in order to avoid superficial friction in large disk areas. The actual construction of this type of turbine is so well known as to require only passing comment here. By reference to Fig. 364, which shows a typical section of a Westinghouse-Parsons turbine, it will be seen that the steam volume progressively increases from inlet *A* to exhaust *B* in the annular space between stator and rotor. The entire expansion, which is approximately adiabatic, is carried out within this annular compartment which essentially corresponds to a simple steam nozzle. There is this difference, however, that whereas in a nozzle the heat energy of the entering steam is expended upon itself in producing high velocities of efflux; in the Parsons turbine the total velocity, due to expansion, is subdivided into a number of steps, in each of which it is reduced through the dynamic relation of jet and vane, so that a comparatively low velocity is maintained from inlet to exhaust; this generally varying from 150 feet per second as a minimum at the high-pressure end to about 600 feet per second as a maximum at the low-pressure end. The action of the steam in this turbine differs from other types also in this respect, that the steam expands in the ring (2) of moving blades (see Fig. 365, so that a reactive effect is produced in addition to the impulse of the steam from (1). The total torque produced at the shaft from ring (2) of moving blades is, therefore, due to impact of steam from (1) and reaction from (2). This process is repeated in each element of the turbine, and the average velocity may be maintained at a uniformly low figure throughout. It is evident that here frictional losses, due to high velocities of efflux, are largely reduced.

34. It has been often held that unless very high vacuum—in fact, an almost uncommercial one—be provided, the economy of the turbine will suffer. This may be so in certain types in which there are idle portions of the bucket wheels rotating in dense media. In the parallel-flow type, however, losses from this source are not so much in evidence, by reason of the fact that the steam is confined to the annulus, and the entire circumference is active in producing torque, thus reducing the proportion of friction to useful work. By reference to the appended tables of tests,

it will be found that the results from turbines operating under poor vacua are not less excellent, relatively, than those obtained with high vacua.

35. The question is frequently asked why three diameters of barrel have been generally selected in the Westinghouse-Parsons turbine. This selection has no bearing whatsoever on the design of the machine, but is merely one of mechanical convenience. The proper expansion can be provided for just as well should there be one or several different diameters. It would be found, however, that if one were used and a speed and diameter of drum were selected that would permit convenient proportions of blades at the outlet, the blades at the inlet of the turbin would become unmechanically small: similarly, if diameters and speeds were

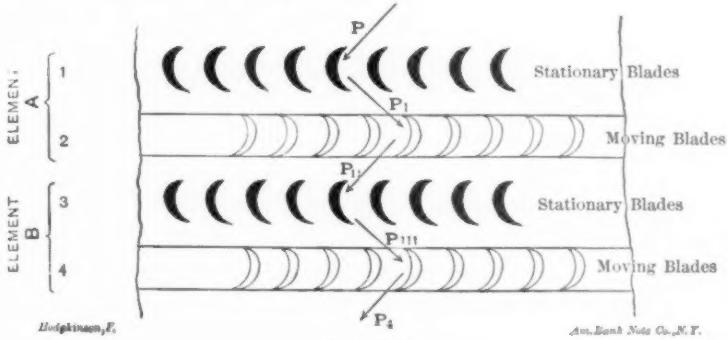


FIG. 365.

selected to suit the inlet blades, the areas of the blades in the last stages would become unmanageably large. By varying the barrel diameters at several convenient points, corresponding variations may be made in the velocity of the steam, thus permitting blade designs of convenient proportions for both extremes.

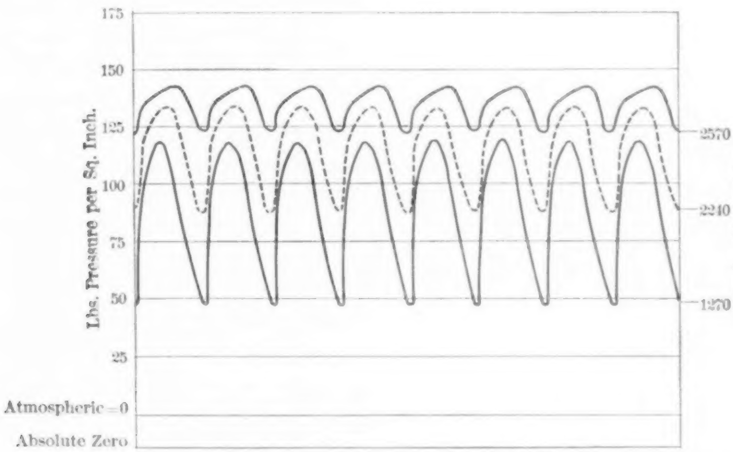
36. It is an important feature of the parallel-flow turbine that the entire annulus between rotor and stator is filled with working steam. It permits the use of large axial clearances between moving and stationary blades without loss in efficiency. In actual practice, this is never less than $\frac{1}{8}$ inch, and in large blades it is as much as 1 inch. In all forms of impulse turbine, separation of nozzle and vane results in surface friction of the jet, and particularly an entrainment of the surrounding medium, as in the fashion of a steam injector, thus increasing fluid friction by drawing the steam backwards through the idle portion of the wheel.

37. Although small axial clearances are unnecessary in the Parsons turbine, it is desirable to employ as small radial clearances as possible in order to prevent leakage of steam from stage to stage. In order to avoid over-estimating the probable extent of this leakage, it is necessary to bear in mind a point which is usually lost sight of, and which, in a considerable measure, offsets the loss from this source. In a machine of given size, the radial clearances between the ends of blades and the walls of the turbine would presumably be constant. The greater leakage would, therefore, naturally occur at the high-pressure end of the turbine, or at the begining of the expansion. By the time the lower stages of the turbine have been reached, the total volume of the steam has become so great, compared with the clearance area, that the latter becomes unimportant. All leakage steam returns energy to the working steam in the form of heat, as its action is similar to wire drawing in a restricted passage; hence it is superheated to a slight degree and serves to partially dry the working steam which contains considerable moisture, due to adiabatic expansion.

38. In any type of turbine it is necessary to provide glands at the ends of the casings to prevent the escape of steam or the influx of air into the turbine at the point of entry of the shaft. Air leakage is particularly detrimental in cases where it is desirable to maintain high vacuum. Various forms of packing glands have been used, but the later type Westinghouse-Parsons turbines are fitted with an arrangement of water-sealed glands. They require no lubrication, and it is impossible for any oil to escape from the bearings or the lubricating system into the steam spaces. There are no rubbing surfaces in these glands, and it is found that they do not wear out. The water used for sealing them is small in quantity, but it is not necessarily lost, as it may, in a power plant, be taken from the feed-pump delivery and the overflow returned to the feed-pump suction.

39. In the longitudinal section of the turbine, Fig. 364, steam is shown entering at S, where a steam strainer is provided, thence through a poppet valve V which is controlled by the governor. When in operation, this poppet valve is continually opening and closing at constant intervals, the periods of which are proportional to the speed of the turbine. Its operation is well shown in Fig. 366, which represents some indicator cards taken on a 1,250 kilowatt turbine at various loads, the indicator being attached to the admission port A, and the indicator barrel revolved at constant

speed by suitable means. At light loads, the valve opens for very short periods and remains closed during the greater part of the interval. As the load increases, the valve remains longer open, until finally continuous full pressure is obtained in the high-pressure end of the turbine. At this time the valve does not reach the seat at all, but is merely vibrating without sensibly reducing the pressure of steam in the turbine. This, in a turbine, would correspond to somewhat over full load.



INDICATOR CARDS SHOWING INITIAL PRESSURES IN A WESTINGHOUSE-PARSONS STEAM TURBINE.

Hodgkinson, F. *Am. Bank Note Co., N.Y.*

FIG. 366.

40. **On the load** being still further increased, an auxiliary or "secondary" valve, designated V, begins to open and admits steam as may be required to a later stage in the turbine where the working steam areas are greater, thus increasing in proportion the total power of the turbine. The operation of this poppet valve is the same as the main admision, so that the governor automatically controls the power and speed of the turbine from no load to such overloads as are generally beyond the limits of generating apparatus built on normal ratings. Its performance may be seen in Fig. 367, which shows an economy test on a turbine where tests were made up to 76 per cent. overload. The economy, it will be observed, drops off slowly as the "secondary" valve opens, but, on the other hand, the range of load at which the turbine may be economically operated is greatly extended, in this case from 400

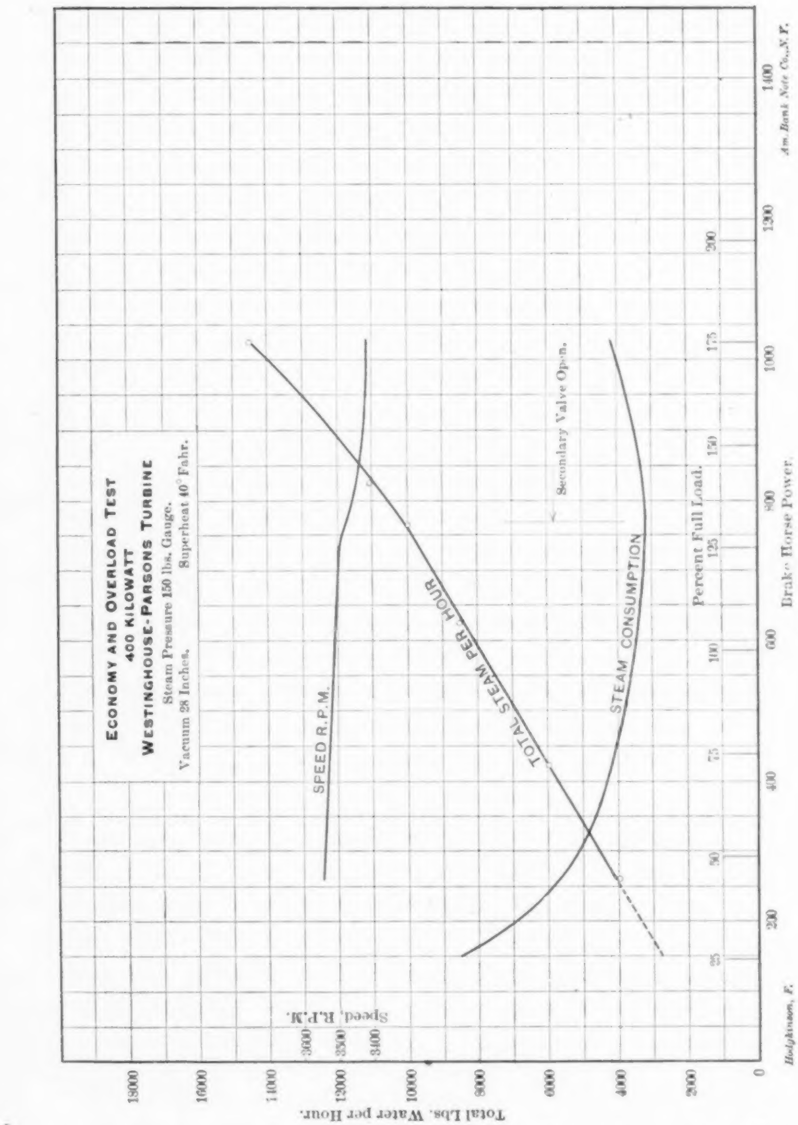

ECONOMY AND OVERLOAD TEST
400 KILOWATT
WESTINGHOUSE-PARSONS TURBINE
Steam Pressure 150 lbs. Gauge.
Vacuum 28 Inches. Superheat 40° Fahr.
SPEED R.P.M.
TOTAL STEAM PER HOUR
STEAM CONSUMPTION
Secondary Valve Open.
Percent Full Load.
Brake Horse Power.
Speed, R.P.M.
Total Lbs. Water per Hour.
Lbs. Steam per Brake Horse Power Hour.
Hodgkinson, F.
Am. Bank Note Co., N.Y.

FIG. 367.

to over 1,000 B. H. P. with a steam consumption varying from 13.2 to 14.3, or but slightly over 8 per cent. from that of maximum efficiency.

41. The intermittent admission of steam to the turbine is productive of some gain in economy at light loads by keeping the temperature range greater than would be possible if the steam were throttled. Another advantage to be derived from this method of admitting steam is that the admission valve and the mechanism that operates it are constantly reciprocating, and consequently get no opportunity to become stuck. The reciprocating motion necessary to operate the mechanism originates with an eccentric driven by the turbine, and is transmitted through the clutch of the governor, causing a continual disturbance; such that the governor is at all times ready to go to a new position with the least change of speed, simply because the "friction of rest" does not have to be overcome. On the larger size turbines, the governor is supplemented by a special automatic centrifugal safety stop which is mounted at the end of the shaft and which actuates, by means of high-pressure steam, an auxiliary self-closing throttle valve located in the main steam pipe supplying the turbine. The safety stop may be set at any predetermined speed, which, if attained, causes the turbine to be brought to rest. It is employed mainly as a precaution against damage due to any possible derangement of the governor mechanism.

42. With the form of governor employed, the speed regulation may be kept to within 2 per cent. between friction load and full load, or 1 per cent. either side of the mean speed. Full load or overloads may be entirely thrown on or off without causing more disturbance than a momentary surge of speed of about 4 per cent. or 5 per cent.

43. The function of the balancing pistons shown at *P*, Fig. 364, is to neutralize the unbalanced axial thrust resulting from the pressure of the steam acting on the various drums of the turbine. These are equal in area to the effective area of the turbine, and are subjected to the same pressure by means of equalizing ports or pipes, *E*. It is evident that, whatever may be the distribution of pressures within the turbine due to varying loads, the thrust in the direction of the pistons must be equal to that in the direction of the blades, with the result that the rotor remains practically in equilibrium. In order, however, to preserve the adjustment of these balance pistons, a thrust bearing of small dimensions is

provided at the end of the shaft, as shown at *T*. This, however, should be called an alignment bearing rather than a thrust bearing, for the reason that, in practice, it takes no thrust. The balancing pistons revolve within the casings with a close fit, but without mechanical friction, and their peripheries are deeply serrated, in order to interpose a path for steam attempting to leak past the pistons, sufficiently devious to render loss from this source as small as possible.

44. A turbine has an important advantage; that local temperature conditions vary but little during operation and on steady load are absolutely constant. The reversals of temperature in a reciprocating engine cylinder have no equivalent. Thus the temperature of rotor and stator is at all points approximately equal to that of the steam in the corresponding expansion stages. Generally the stator is made of cast iron and the rotor of steel, so that the differential expansion that must necessarily exist between them has the effect of increasing or decreasing the axial clearances between runing blades at different loads. This, however, is unimportant with this type of turbine because of the ample axial clearances provided. The exhaust end of the turbine is bolted to the bedplate, while the steam end is provided with a sliding foot working between machined ways on the bedplate, so as to permit the turbine to expand, as it will. In types of turbines, however, where small axial clearances exist, this question of differential expansion is not such a simple matter.

45. It has been advanced that the blade construction employed in the Parsons turbine constitutes an element of complexity, which also is not conducive to low cost of construction. The author is, perhaps, not conveying new information in stating that the blades are rolled out from special bronze or steel into long strips, then sawed into the proper length, and finally mounted around the periphery of the rotor and the stator in grooves with special separating pieces, the whole being finally caulked in position. In practice, it is found that with this construction the blades are never released, except through some special cause, and the construction is of immense advantage in minimizing the delays due to accidents.* A turbine opened up for inspection is shown in

---

* Such accidents are, however, of rare occurrence, and one that may be attributed to the blading construction has yet to happen. On one occasion an expanding exhaust pipe, which had been too firmly anchored at the lower end, occasioned sufficient distortion of the turbine casing to destroy several rows of blades. The

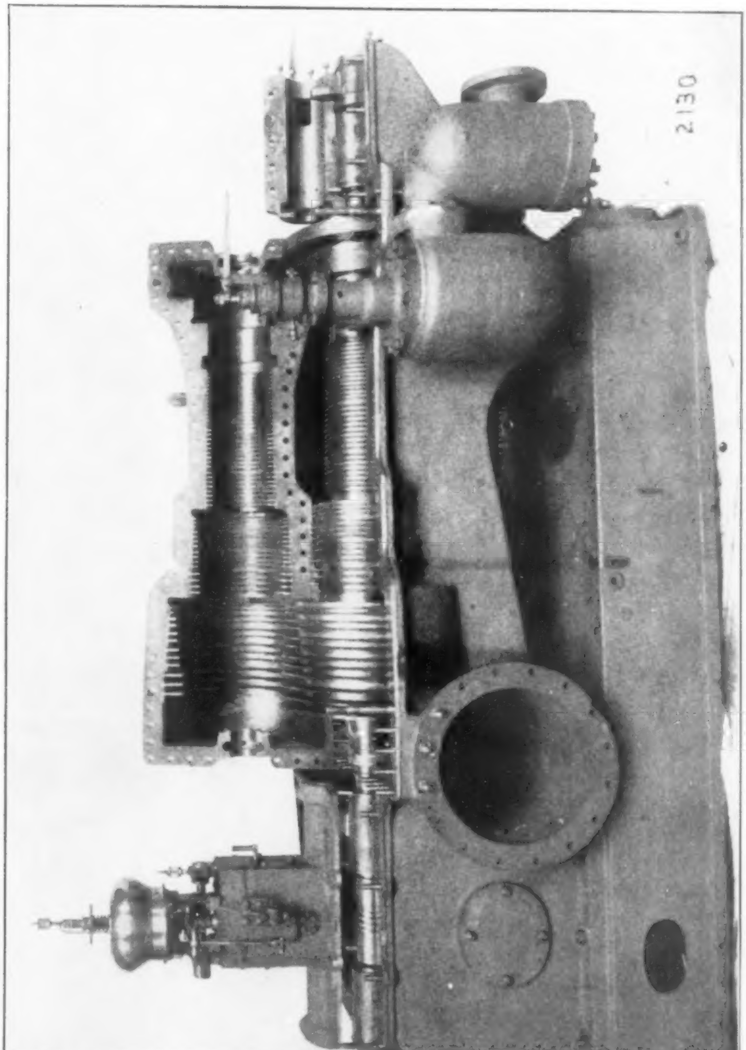


FIG. 368.—A 400-K. W. TURBINE OPEN FOR INSPECTION.

Fig. 368. By placing the shaft in a sling, the rotor can be lifted out by a crane and the entire interior examined. Such a turbine can be taken apart and completely reassembled inside of two hours, which cannot be said of many types of prime movers now known to us.

46. In view of the widespread discussion upon the merits of a reheater in reciprocating engine work, it may be of interest to mention the results of its application to steam turbines. The earlier Westinghouse-Parsons turbines of large size were constructed in tandem two-cylinder form for the purpose, not only of reducing the span between bearings, but also to permit the introduction of a receiver between high-pressure and low-pressure cylinders, the arrangement being intended to separate as much moisture as possible resulting from adiabatic expansion, and to superheat the remaining steam. Exhaustive tests have, however, shown the reheater to be of little, if any, value in increasing the economy of the turbine when the high-pressure steam condensed in the reheater coils was charged up against the turbine. An improvement in the separator resulted in an improvement in the operation of the reheater, but, notwithstanding this, no advantage due to the reheater could be observed, and its application does not seem warranted, on account of the decreased compactness of the machine.

47. In the course of regular operation of power stations, it is not an unusual occurrence that wet steam comes over from the boilers, due to foaming, overfilling, or other causes. The effects upon reciprocating engine machinery need not be commented upon here. In several instances of turbine plants, slugs of water coming over from the boilers have been known to bring the turbine almost to a standstill, but without any apparent damage resulting. As soon as the water passes over to the condenser, the turbine again regains its speed. The effect on the economy of entrained moisture in the steam has been found to increase the steam consumption to an amount about twice the percentage of moisture in

---

turbine was immediately shut down, the casing opened and the debris removed. It was then again put under steam and continued in full load service during the day without further trouble or apparent effect on its capacity. At night, after extra blading had arrived from the factory, the machine was again opened and the damaged rows replaced. The repairs were made during several short stoppages at nights, but the turbine was kept in uninterrupted daily operation. The accident only kept the turbine out of service about three hours.

the steam; *i.e.*, 2 per cent. of moisture will decrease the economy about 4 per cent.

48. Finally may be mentioned the system of lubrication desirable with turbine outfits. Essentially a point in mechanical design and independent of the steam cycle, it is nevertheless one on which the successful operation of any machine is absolutely dependent. Although possessing extraordinary features of excellence in the matter of securing great compactness of design, forced lubrication, as ordinarily understood to mean—oil under high pressure—does not seem entirely desirable. In the Parsons type of turbine, the projected areas of the journals are proportioned so that the entire weight of the rotating element may be supported upon a fluid film of oil through capillary action alone. A small pump, driven from a worm gear upon the shaft, circulates oil through a closed system, comprising in the order of their arrangement: pump, oil cooler, bearings, and reservoir. The oil is always applied to the bearings at the point of least pressure; that is, it enters at one end and follows a groove along the top of the shell, from which it is distributed around the shaft. The pressure impressed upon the fluid films is due simply to a static head of 1 to 3 feet, sufficient to insure thorough flushing of the bearings. It is probable that the shaft never comes into actual contact with the bearings, but is separated by the oil film. This is evident by the fact that bearings when taken out for inspection after several years' continuous run are found to be subject to practically no wear. In several cases, the original tool marks upon the interior of the shell have been preserved.

49. The benefits resulting from the employment of a closed oil circulating system and large bearing areas are apparent in the cost of operating a turbine in regular service. As a result of inquiries in several concerns employing Westinghouse-Parsons turbine machinery, it has been elicited that a turbine ordinarily consumes about $\frac{1}{4}$ gallon of high-grade engine oil per kilowatt capacity per year; or, in other words, the total quantity of oil used per year averages about 100 gallons for a 400 kilowatt turbine. As this oil costs from 25 to 50 cents a gallon, the expense for oil for the turbine is not ordinarily over $7\frac{1}{2}$ to $12\frac{1}{2}$ cents a day, and even this is not all directly chargeable to the turbine, as it is common practice to utilize the oil which is removed from the circulating system on auxiliaries and other low-speed machinery.

50. It is well known that flexible bearings are employed on

Parsons turbines of small sizes in order to permit the rotor to revolve about its gravity instead of its geometric axis, this being necesary at the high speeds employed in order to neutralize the effect of minute errors in the balancing of the disks. The flexible bearings consist of a nest of concentric bronze sleeves with sufficient clearance between them to permit the formation of oil films, which act as cushions, permitting a certain amount of vibration of the shaft, but at the same time restraining such vibration within narrow limits. In the larger sizes of turbines, however, and, in fact, for all machines running below 1,200 revolutions per minute, the flexible bearing is no longer found necessary, and is replaced by a solid split self-aligning journal, lined with anti-friction metal, as in the ordinary forms of low-speed machinery.

Turbine Generators:

51. It is an interesting fact that, owing to the introduction of steam turbines, the general characteristics of generating apparatus have been modified to a wide extent, and in points of running speeds have returned to the practice of the first builders of electrical machinery. Owing to the restrictions placed upon the designers by reciprocating engine speeds, the dimensions and bulk of engine type generating machinery have, of late years, become enormously increased: similarly, the cost of construction. With the advent of the turbine, however, speeds have been increased to such a point as to secure in the generator construction minimum bulk and cost consistent with strength and durability.*

52. The turbine generator is more easily applied to alternating current work, for the reason that commutation difficulties involved in direct current machinery running at high speeds are avoided. The preferable construction, therefore, comprises rotating field and stationary armature. In present turbine generators the armature construction is not essentially different from that of the ordinary engine type machines. In the construction of the field, however, the centrifugal stresses necesitate a con-

* A pertinent comparison may be made in the two types of 5,000 kilowatts generators which will form the power equipment of the Rapid Transit Subway in New York City. The engine type generators run at 75 revolutions per minute, are approximately 40 feet in diameter and weigh 980,000 pounds. The turbine generators, on the other hand, run at 750 revolutions per minute, are 13 feet, 6 inches in diameter, and weigh 234,000 pounds, the weight of journals and shaft excluded in each case. The engine type generators have 40 poles, and the turbo-generators four, giving the same frequency—25 cycles per second.

struction of greater inherent strength. Recent practice embraces two designs—one of built up form—used in fields having six or more poles, and the other of a solid steel casting thoroughly annealed, bored for the reception of the shaft, and slotted axially for the reception of bar or strap windings which are insulated and confined in position by wedges.

53. The turbine, however, makes possible the use of a still further type of generator, which, although presenting difficulties in design at ordinary engine speeds, becomes ideally suited for direct connection to the turbine, both by reason of its electrical characteristics and its inherent strength of mechanical construction. It is well known that, if the ordinary squirrel cage induction motor runs below synchronism with the system upon which it is operated, it will absorb power from that system proportionate to the slip or drop in speed. If it is run in synchronism therewith by external means, it will absorb no power; and if run above synchronism, it will become a generator and return electric power to the system.

54. When running below synchronism, the greater part of the current absorbed by the motor appears as power, but a small part is consumed within the motor itself in magnetizing its rotating field. When run above synchronism, the motor, now a generator, still requires magnetizing current from the line to which it is connected. It is, therefore, incapable of operating by itself, and must be run in connection with synchronous machinery capable of supplying its magnetizing current and controlling the frequency of the system.

55. The induction or non-synchronous generator, unfortunately, imposes a lagging curent upon the supply system, but the power factor can be brought within a few per cent. of unity, so that the effect upon the system may be readily neutralized. Its peculiar electrical characteristics impose limitations upon its general use for power station work, but when employed in conjunction with synchronous apparatus, such as ordinary alternators, synchronous motors and rotary converters, it becomes peculiarly suitable for extension to a power system in which the limit of generator capacity has already been reached. With the apparatus mentioned, particularly with synchronous motors and rotary converters, a sufficient leading current may be impressed upon the system by over-exciting the fields of these machines to entirely neutralize the effects of the magnetizing currents required

by the induction generator, so that, in general, if existing apparatus is ample to care for existing inductive loads with reasonable margin, the induction generator can be employed to great advantage.

56. A feature which is particularly favorable in rendering it suitable for turbine driving is, that by largely reducing the number of poles the magnetizing currents may be largely reduced. For this reason, the limitations of the induction generator occur largely in the direction of bulk rather than otherwise. As it must operate at the comparatively high speed of the turbine, it is thus possible to reduce the number of poles to a few pairs, so that the losses above mentioned are minimized and the generator becomes commercially practicable. And as the squirrel cage construction of the rotor is peculiarly well suited for high speed work, we are fortunate in having here one of the few cases in which the electrical and mechanical conditons governing generator and prime mover are almost exactly suited to each other. In general, the higher the speed at which the machines can be safely operated, the less the material necessary and the smaller the losses, resulting in an extraordinarily high efficiency and power factor.

57. For example, with a two-pole 60-cycle induction generator of 500 kilowatts, running at 3,600 revolutions, the power factor may be brought as high as 98 per cent. or higher at full load, and the total efficiency will be far greater than that of present generating machinery.

Economy:

58. As one of the principal claims which the turbine makes is economy of steam, and consequently of fuel, a few observations may be made upon this subject. As is well known, high economy has been obtained with the turbine operating under favorable conditions, but as abstract figures of steam, heat or fuel consumption rarely convey an adequate idea of the actual merits of the prime mover in question, it is necessary to examine the conditions under which these results were obtained.

The table of tests appended hereto has been prepared with a view to presenting in as concise shape as possible, the results of several hundred tests upon turbines of all sizes and under all conditions of operation, the headings being arranged with reference to these conditions. The tests were conducted in the testing department of the Westinghouse Machine Company at East Pittsburg,

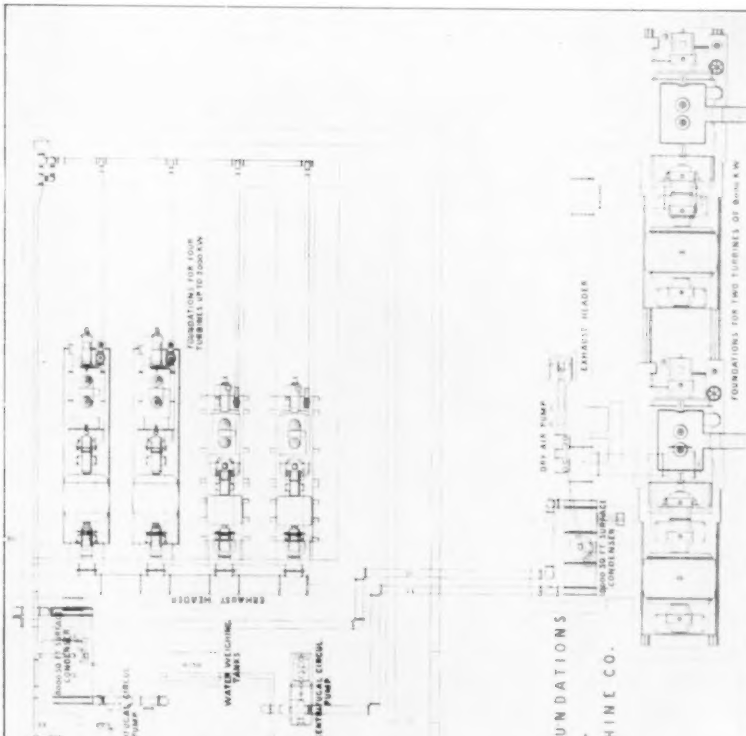

PLAN OF
STEAM TURBINE TESTING FOUNDATIONS
AND CONDENSERS.
THE WESTINGHOUSE MACHINE CO.
WATER RESERVOIR
CONNECTED WITH TURTLE CREEK
WATER WEIGHING TANKS
EXHAUST HEADER
DUPLEX CIRCUL PUMP
CENTRIFUGAL CIRCUL PUMP
DRY AIR PUMP
FOUNDATIONS FOR TWO TURBINES

FIG. 369.

where every turbine put through the shops is thoroughly tested before shipment.

59. Columns 51 and 52 give results under moderately favorable conditions—150 pounds steam pressure, 28 inches vacuum and 180 degrees superheat; and in columns 1 and 3 will be found tests under decidedly unfavorable conditions—125 pounds pressure, 26 inches mercury vacuum and saturated steam.

60. A plan of the testing department is shown in Fig. 369, and provides facilities for the accommodation of

Four turbines of small capacity up to 500 kilowatts.
Four turbines of 2,000 kilowatt capacity.
Two turbines of 5,000 kilowatt capacity.

The latter foundations at this time are not yet complete.

61. This is somewhat of a radical departure in manufacturing methods and entirely unprecedented. At present, it is unusual practice among generator builders to determine efficiency in large machinery by other than the motor-generator method, and in the largest sizes, such as the 5,000 kilowatt Manhattan generators, the machines are not even turned over in the factory; similarly, engines above 500 to 1,000 horse-power are seldom tested in the shop, and the larger sizes are shipped without having steam turned into them. It is thus of peculiar interest that the largest turbine units will be tested under steam at the shops. The testing equipment comprises boiler plant, a gas-fired superheater and four independent surface condenser outfits, ranging in size from 1,600 square feet up to 10,000 square feet surface.

62. The condensers, with the exception of the smallest, are all of the "counter current" type, exhaust steam being admitted from beneath. The condensed water is received in a hot well located below the condensers, and air is withdrawn from the top by two stage dry vacuum pumps. These are capable of maintaining a vacuum within half an inch of the barometer, with a closed suction. The vacuum with the condenser in operation depends, of course, upon the temperature condition within the condenser.

63. Tests are made by means of brakes or by electric generators as desired. For the latter, large water rheostats are available, while for the former, a special form of water friction brake has been devised which has proven extremely flexible in its application, and of great value.

64. Steam consumption is determined by weighing condensation in the usual manner. Vacuum readings are all reduced to a

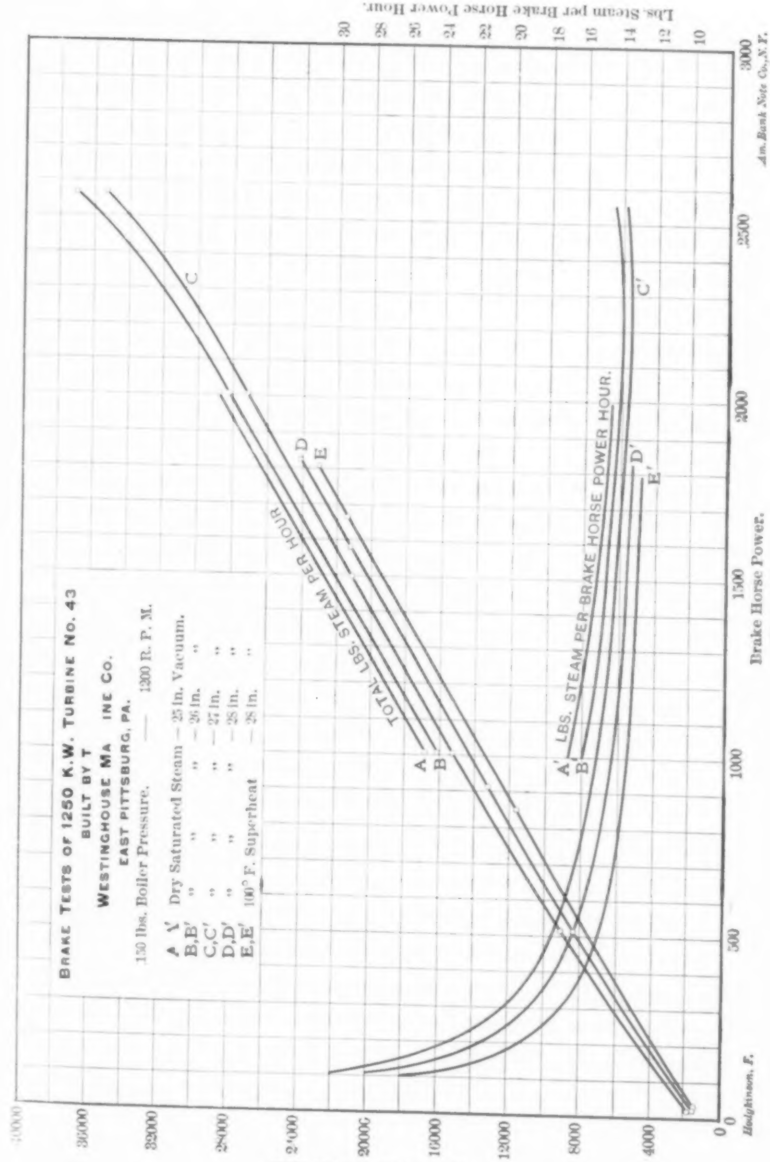

Brake Tests of 1250 K.W. Turbine No. 43
Built by T
Westinghouse Ma ine Co.
East Pittsburg, Pa.
150 lbs. Boiler Pressure. — 1200 R. P. M.
A A' Dry Saturated Steam — 25 in. Vacuum.
B,B' " " — 26 in. "
C,C' " " — 27 in. "
D,D' " " — 28 in. "
E,E' 100° F. Superheat — 28 in. "
TOTAL LBS. STEAM PER HOUR
LBS. STEAM PER BRAKE HORSE POWER HOUR.
Total Lbs. Steam per Hour
Lbs. Steam per Brake Horse Power Hour.
Brake Horse Power.
Hodgkinson, F.
Am. Bank Note Co., N.Y.

Fig. 370.

basis of 30 inches of mercury necessitated by the elevation of Pittsburg.

65. The tests shown in columns 38 to 43 of the table, are plotted in Fig. 367. This curve embodies the results of the introduction of the secondary governor valve, and shows a remarkable range of load with high economy.

66. Some brake tests of a 1,250 kilowatt turbine are plotted in Fig. 370, with vacua ranging from 25 inches to 28 inches, and clearly show the effect of vacuum and superheat.

67. Fig. 371 and columns 93 to 100 show some electrical tests on a similar machine, these having been verified by Mr. Julian Kennedy, Consulting Engineer of Pittsburg.

68. Fig. 372 shows the plotted results of tests carried out at the Westinghouse Machine Company's works by Mr. F. W. Dean of the firm of Dean & Main, and shows a very good performance for a small machine.

69. The general appearance of these curves might possibly give the impression of a poor economy at lighter loads, especially to engineers who have been accustomed to considering engine performance on a basis of indicated horse-power. When they consider such performance on a basis of brake or electrical horse-power, they readily make a mental correction between brake or electrical horse-power and indicated horse-power at full load, but seldom realize the fact that the mechanical efficiency is much poorer at fractional loads than at full loads.

70. It may be said that the mechanical losses of an engine are approximately constant at all loads, and assuming this, an engine that has 94 per cent. mechanical efficiency at full load, has an efficiency of but 88.6 per cent at half load, and at quarter load of 79.8 per cent. To exhibit this, Fig. 373 has been prepared with two of the tests already shown in Fig. 372 plotted again with the curves *C* and *D* added. The method of plotting curves *C* and *D* has been as follows: Take, for instance, curve *C*:

From tests, brake horse-power at rated full load .... = 593.17

Internal horse-power $= \dfrac{593.17}{.94}$ .... = 631.03

Loss, horse-power 37.86

This loss has been assumed constant at all loads.

Total steam, pounds per hour .... 8,249.

Pounds steam per indicated horse-power hour .... $\dfrac{8{,}249}{631.03} = 13.08$

Pounds steam per indicated horse-power hour, when doing, say, 300 brake horse-power .... $\dfrac{4{,}610}{300 + 37.86} = 13.66$

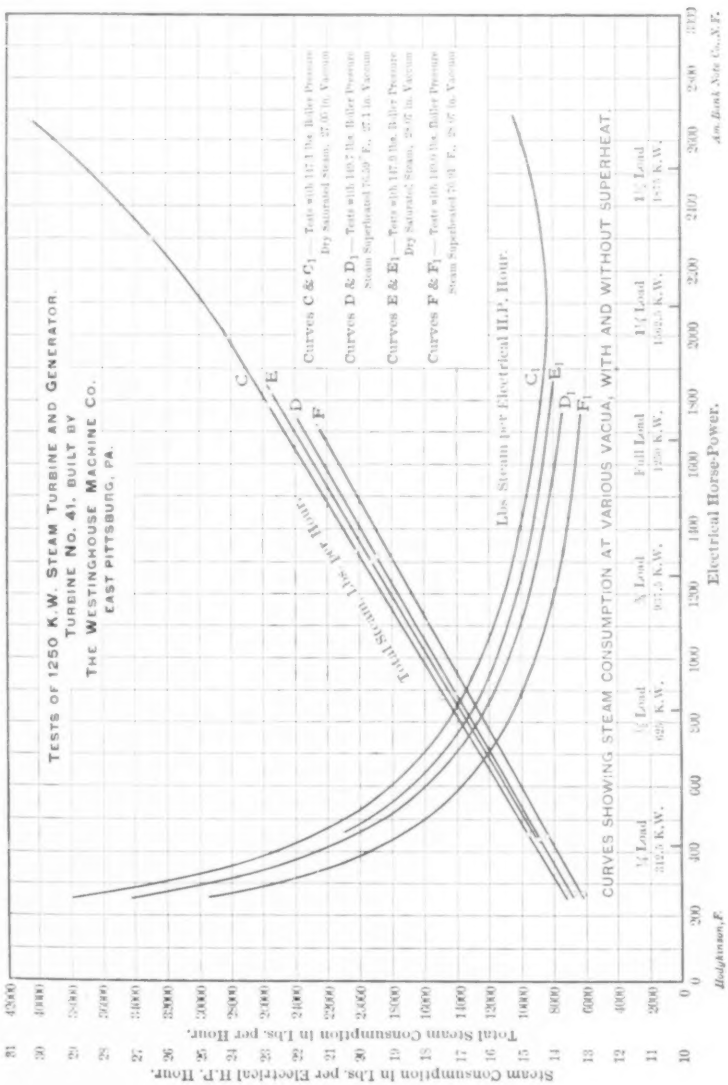

TESTS OF 1250 K.W. STEAM TURBINE AND GENERATOR.
TURBINE NO. 41. BUILT BY
THE WESTINGHOUSE MACHINE CO.
EAST PITTSBURG, PA.
CURVES SHOWING STEAM CONSUMPTION AT VARIOUS VACUA, WITH AND WITHOUT SUPERHEAT.
Total Steam, Lbs. per Hour.
Lbs. Steam per Electrical H.P. Hour.
Electrical Horse-Power.
Total Steam Consumption in Lbs. per Hour.
Steam Consumption in Lbs. per Electrical H.P. Hour.
Full Load
1250 K.W.

FIG. 371.

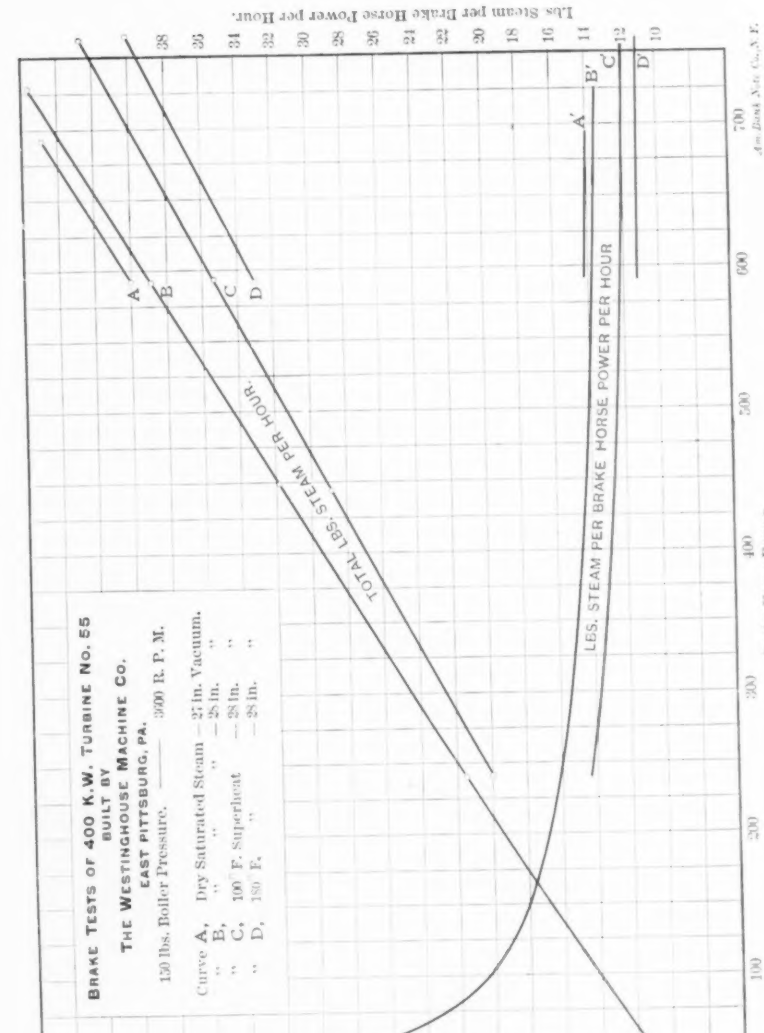

BRAKE TESTS OF 400 K.W. TURBINE No. 55
BUILT BY
THE WESTINGHOUSE MACHINE CO.
EAST PITTSBURG, PA.
150 lbs. Boiler Pressure. —— 3600 R. P. M.
Curve A, Dry Saturated Steam — 27 in. Vacuum.
" B, " " — 28 in. "
" C, 100° F. Superheat — 28 in. "
" D, 180° F. " — 28 in. "
TOTAL LBS. STEAM PER HOUR.
LBS. STEAM PER BRAKE HORSE POWER PER HOUR
Total Lbs. Steam per Hour
Lbs. Steam per Brake Horse Power per Hour.
Brake Horse Power.

FIG. 372.

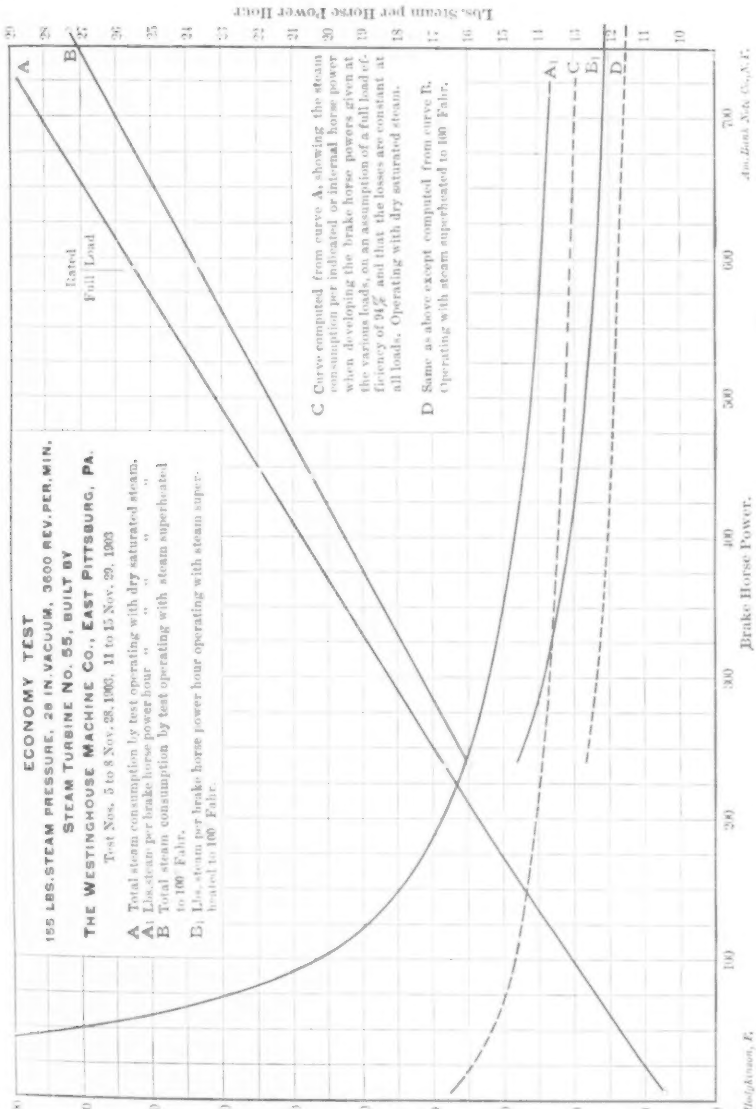

ECONOMY TEST
155 LBS. STEAM PRESSURE, 28 IN. VACUUM, 3600 REV. PER. MIN.
STEAM TURBINE NO. 55, BUILT BY
THE WESTINGHOUSE MACHINE CO., EAST PITTSBURG, PA.
Test Nos. 5 to 8 Nov. 28, 1903, 11 to 15 Nov. 29, 1903
A Total steam consumption by test operating with dry saturated steam.
A1 Lbs. steam per brake horse power hour " " " "
B Total steam consumption by test operating with steam superheated to 100° Fahr.
B1 Lbs. steam per brake horse power hour operating with steam superheated to 100° Fahr.
C Curve computed from curve A, showing the steam consumption per indicated or internal horse power when developing the brake horse powers given at the various loads, on an assumption of a full load efficiency of 94% and that the losses are constant at all loads. Operating with dry saturated steam.
D Same as above except computed from curve B. Operating with steam superheated to 100 Fahr.
Rated Full Load
Total Lbs. Steam per Hour.
Lbs. Steam per Horse Power Hour
Brake Horse Power.

FIG. 373.

In this way curves *C* and *D* show an indicated horse-power performance at light loads that is particularly good.

71. In columns 122, 123 and 124, are shown some tests on a 2,600 kilowatt turbine tested under high operating conditions. The turbine, however, is of the Parsons type, and was built by Brown-Boveri & Company for the Municipal Electric Power Station for Frankfurt-on-Main.

72. From these curves and table of tests may be gleaned a number of interesting facts which it may be worth while to point out here.

73. *First.* The Willans line or curve of total water consumption is approximately a straight line at all points up to the opening of the secondary governor valve on heavy overloads. This relation has an immediate thermodynamic meaning and points to the utilization of steam in the turbine with the same internal efficiency at all loads; or, in other words, that the losses in the turbine from all causes, thermal, thermodynamic and mechanical, are approximately constant at all loads.

74. *Second.* The necessity of high vacua and high superheat is not essential to high economy, as has been before alluded to. This is shown in tests of a 400 kilowatt turbine under 26 inch vacuum, 125 pounds pressure and saturated steam. A water rate of 15.41 pounds per B. H. P. was obtained, which, although not remarkable, would seem to bear out the supposition of small fluid frictional losses within the turbine. Another result of 14.4 pounds steam per B. H. P. hour, obtained with a 1,250 kilowatt turbine operating with 150 pounds boiler pressure and 25 inches vacuum is of interest.

75. *Third.* The gradual improvement in economy with an improvement in operating conditions is well brought out by the tests on the 1,250 kilowatt turbine. See Figs. 370 and 371. By increasing the vacuum from 27 inches to 28 inches, and the temperature of the steam from that corresponding to dry saturation to 77 degrees Fahr. superheat, the full load steam consumption was reduced from 14.6 pounds to 13.2 pounds per E. H. P. hour.

Foundations and Power Plant Designs:

76. With steam turbines, practically no foundations are necessary, merely something that will uphold the dead weight of the machine. Foundation bolts are never used except on shipboard. Operation of turbines on light flooring is entirely satisfactory,

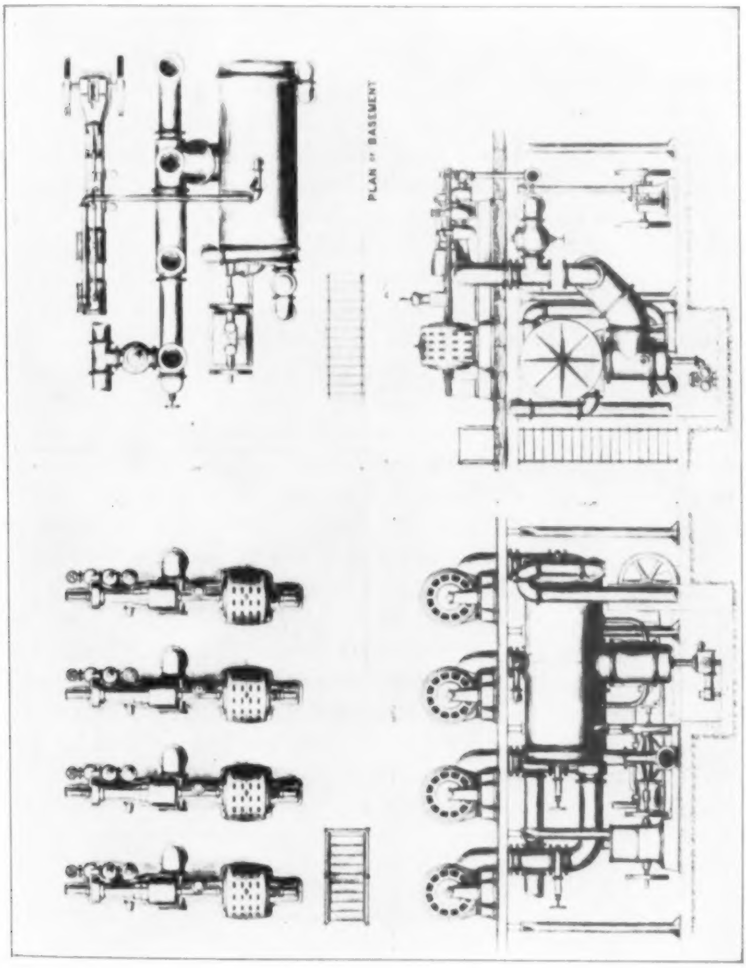


FIG. 374.—ENGINE ROOM PLAN FOR FOUR 400-K. W. STEAM TURBINES.

thus permitting their being placed on upper floors of buildings. This also permits of the condensing plant being located immediately below the turbine, by this means the total plant occupying the minimum amount of floor space.

This is the construction made use of in the new power house of the Westinghouse Electric and Manufacturing Company.

77. Some condenser layouts have been prepared embodying the above principles. Fig. 374 shows an engine-room layout for four 400 kilowatt turbines, all exhausting into a central surface condenser of 7,000 square feet cooling surface. The condenser equipment consists of a dry vacuum pump, circulating pump and condensed water pump, and is suitable for maintaining 28-inch vacuum. Allowing ample space for passageways, etc., the engine-room covers a space of 35 feet by 26 feet. The basement is 14.6 feet deep. Turbines are placed at 7 feet 10-inch centres.

78. A similar layout, shown in Fig. 375, embraces four 1,000 kilowatt turbine generators. In this case, the condenser equipment consists of a surface condenser of 4,000 square feet surface, a circulating pump and a condensed water pump for each turbine. Two air pumps are shown, either one of which is big enough to take care of the whole plant. In this case, the engine-room occupies a space of 59 feet by 36 feet, the basement being 18 feet deep, turbines being placed at 13 feet centers.

79. A larger engine-room layout is similarly shown in Fig. 376, consisting of four 5,500 kilowatt turbines. Here each turbine is equipped with a complete separate condensing outfit, the condensers each having 20,000 square feet surface. The engine-room occupies 100 feet by 61 feet, and the basement is 23 feet deep. The turbines are placed at 22 feet 6 inch centers.

Tabulating the above figures, we have

Number of Units.	Normal capacity of each unit K.W.	Normal capacity of Engine Room K.W.	Square feet area of engine room.	K.W. capacity per square foot of engine room.	Square foot of engine room per E.H.P.
4	400	1,600	910	1.76	.424
4	1,000	4,000	2,124	1.88	.396
4	5,500	22,000	6,100	3.60	.207

It will be observed that means are provided in all these cases for operating any one of the turbines non-condensing.

In all these layouts surface condensers have been shown, be-

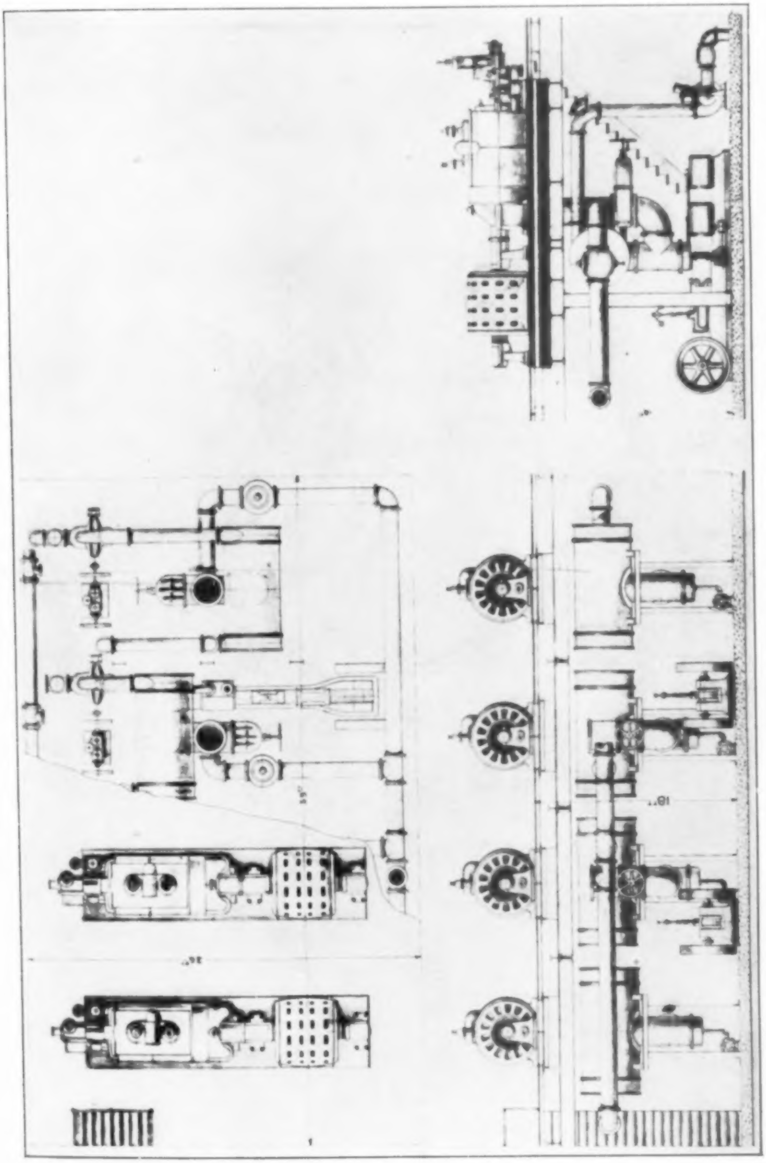


Fig. 375.—Engine-room Plan for Four 1,000-K. W. Steam Turbines

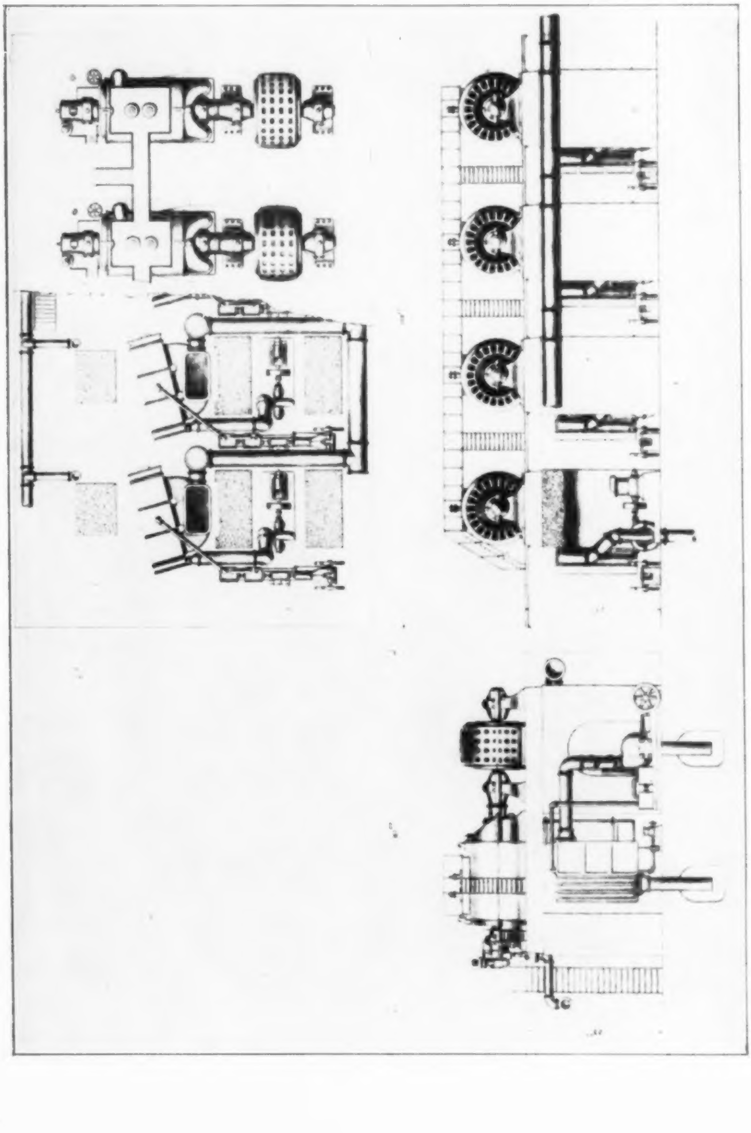


Fig. 376.—Engine-room Plan for Four 5,500-K. W. Steam Turbines.

cause it is presumable surface condensers will be more frequently employed in connection with turbines than other types, if only because of the advantage of absolutely clean feed water.

80. It is claimed by condenser builders that with modern dry vacuum pumps, and a closed hot well system, that a better vacuum can be obtained than with a jet condenser, due to the fact that the feed water does not become aërated.

81. The cost of operating a surface condenser can, under favorable conditons, be kept very small, especially when the circulating water does not have to be raised to any height.

82. A jet condenser, on the other hand, has to do work in order to expel the cooling water against nearly an atmospheric pressure, according to the vacuum; but is a considerably simpler piece of apparatus is less costly and is generally not subject to electrolytic troubles that are sometimes incidental to surface condensers.

83. Barometric condensers make a very suitable type of condenser for large vertical engines, especially when the steam can get direct to the condenser without having to be carried upward.

84. With turbines, in connection with this type of condenser, the author has observed that some work is required to carry the water in the exhaust steam to the top of the condenser.

85. In one instance this was carefully observed in connection with a 1,500 kilowatt turbine. The exhaust left the turbine cylinder at the bottom by means of two elbows and about six feet of horizontal pipe, passing up a vertical pipe to the condenser. Except on fairly heavy loads, the horizontal pipe had water in it, observed by a gauge. The amount of this water may be said to have been a measure of the load. The back pressure due to this piece of pipe and two elbows, together with the water laying in the bottom, amounted to $\frac{3}{8}$ inch mercury with steady load.

86. If the load became less, the back pressure would disappear until more water collected in the pipe and the same $\frac{3}{8}$ inch back pressure would be reëstablished.

87. If, on the other hand, the load increased, this back pressure would rise sometimes to as high as 1 inch, until the water could be carried away, when it would fall back again to about the same $\frac{3}{8}$ inch.

88. As turbines can expand down to the utmost limits of exhaust pressure—it is desirable to give the turbine every advantage in this respect—hence it is well to avoid carrying the exhaust

up hill, thus giving the water that must necessarily exist in the exhaust, an opportunity to drain and keep the exhaust pipe free.

89. While the author has endeavored to point out that high vacuums are not necessary to the successful operation of steam turbines, the higher economy obtained with high vacuum warrants the condenser problem being carefully considered.

It will be seen by the table of tests, that each inch of vacuum above 26 inches will benefit the economy from 3 per cent. to 4 per cent.

90. Assuming a 1,500 kilowatt turbine, operated at full load for a day at 28 inches vacuum instead of 26 inches vacuum, it will save approximately 1 pound of steam per horse-power per hour, or 48,000 pounds a day of twenty-four hours. If we allow for one pound of coal costing \$2 per ton, and evaporating 7 pounds of water, this will mean a saving of 6,875 pounds of coal, representing a daily saving of \$6.87 or \$2,061 a year. Thus the difference between the two condenser investments, being \$4,000, would accordingly pay interest of about 50 per cent., due to the saving in coal.

91. Such figures as these, however, are of no practical value, because of the time a power plant is running at fractional loads, but nevertheless, it is apparent that it is worth while to employ high-class condensers. A point, however, which should not be lost sight of is, that the higher vacuum gives a greater percentage gain in economy at fractional loads than at full loads. In this matter must be considered the extra cost of operating with a high vacuum.

With air leaks eliminated and a closed hot well system, the air pump should take no more power because of high vacuum.

A dry air pump will obviously be doing no work beyond its own friction when there is no vacuum in the condenser. Similarly, it will be doing no work when there is a perfect vacuum in the condenser, provided there are no air leaks.

It may be interesting to record that the maximum load when the air pump is started comes on when the vacuum is about 20 inches to 21 inches.

92. With the circulating water pump, however, the matter is different, as it will have approximately two or three times as much water to handle, according to its inlet temperature, with the higher vacuum. The power required to do this varies in individual cases, but it often happens that the water can be returned to the same

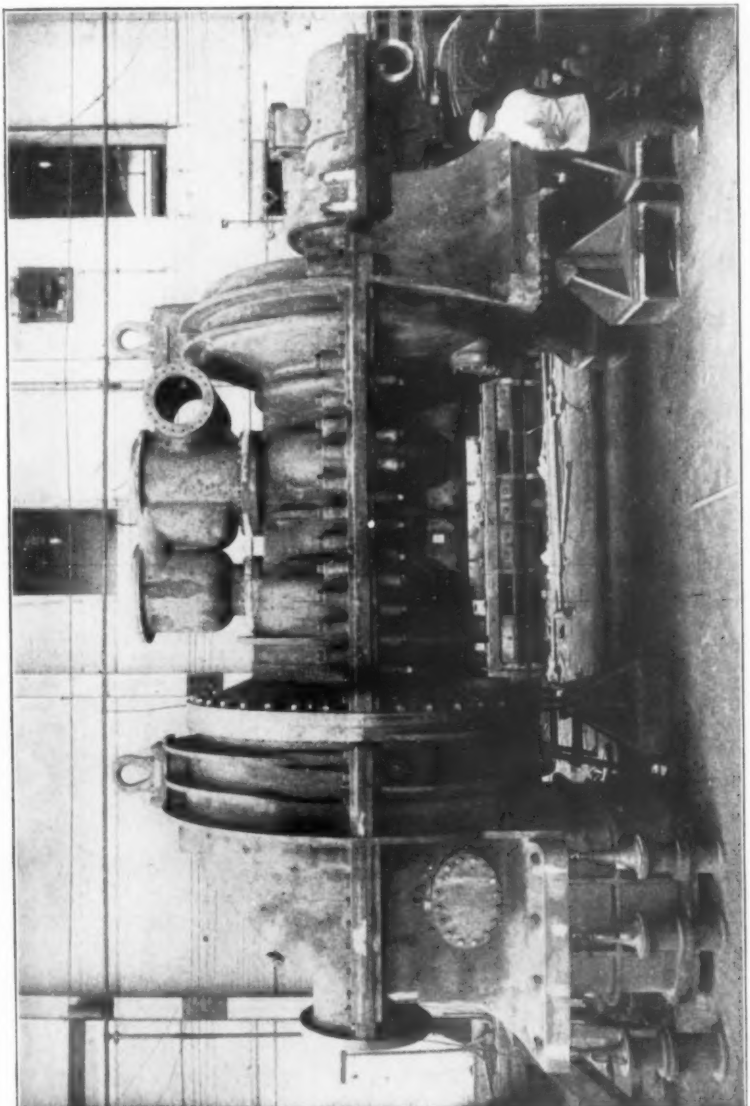


Fig. 377.—5,500-K. W. Westinghouse-Parsons Steam Turbine.

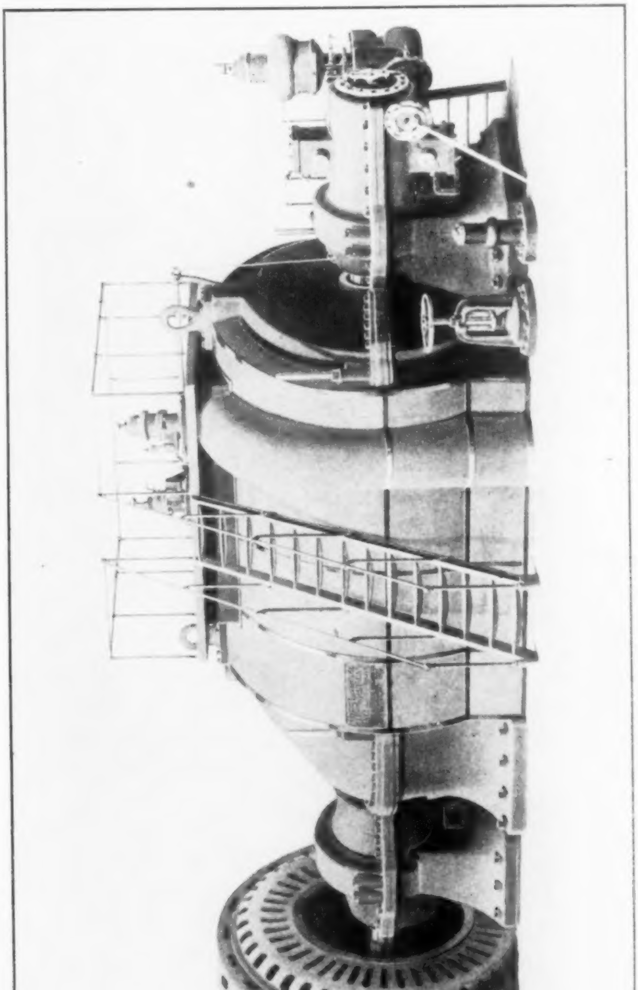


Fig. 378.—5,500-K. W. Westinghouse-Parsons Steam Turbine.

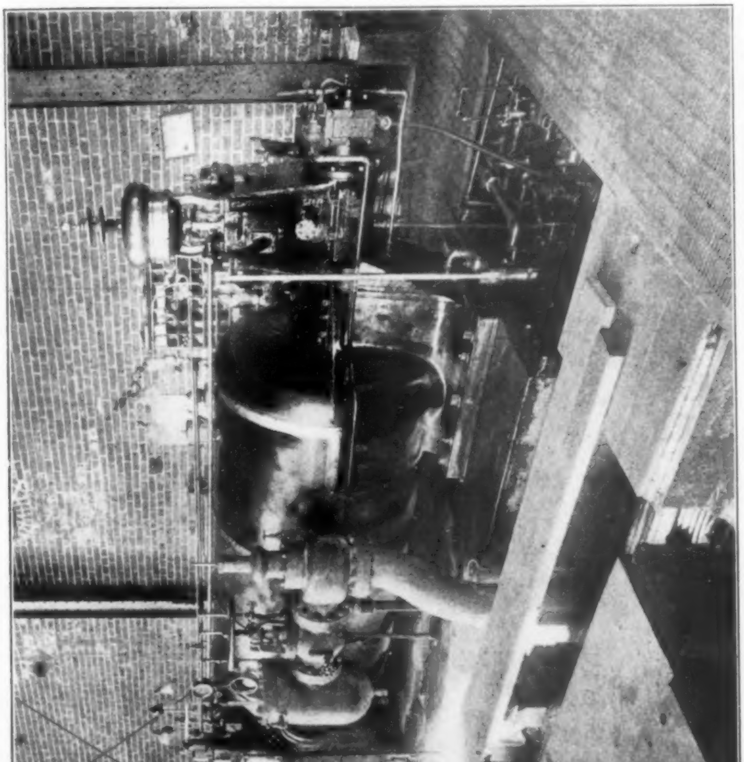


FIG. 379.—WESTINGHOUSE-PARSONS STEAM TURBINE AT CLEVELAND & SOUTHWESTERN TRACTION CO.'S POWER PLANT, AT ELYRIA, O.

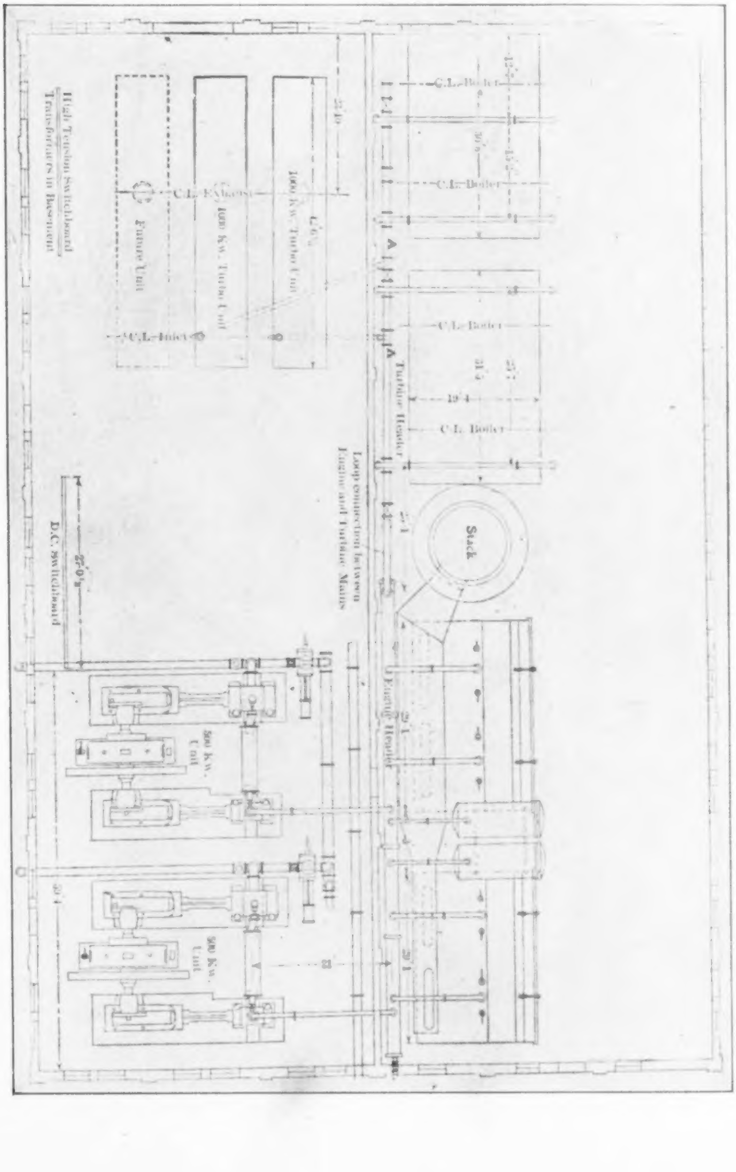


FIG. 380.—ENGINE ROOM PLAN AT CLEVELAND & SOUTHWESTERN TRACTION CO., ELYRIA, OHIO.

level from which it has been taken, such that the circulating water system forms a syphon, and the pump has only the fluid friction of the pipes and condenser tubes to overcome.

93. Fig. 377 shows a 5,500 kilowatt turbine, similar to those referred to above, in course of construction. A good idea of its final appearance is given in Fig. 378. Its overall dimensions, including generator, are 47 feet 3 inches by 16 feet by 14 feet. Maximum overload capacity, 13,000 horse-power. On this basis

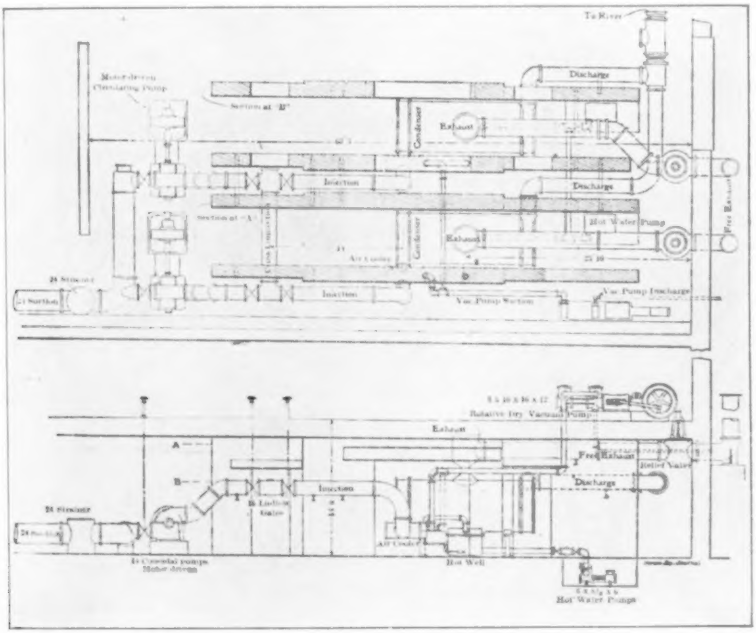


FIG. 381.—PLANS SHOWING CONDENSER ARRANGEMENT, CLEVELAND & SOUTHWESTERN TRACTION CO., ELYRIA, OHIO.

we have over 17 horse-power to a square foot, including the generator. The speed will be 750 revolutions per minute.

In closing it may be appropriate to make passing comment on a few typical turbine installations now in operation.

94. Fig. 379 shows the turbine installation at the Cleveland and Southwestern Traction Company's power house at Elyria. This machine has been successfully in operation since early in November, 1903, at times carrying the whole load of the station amounting to 1,600 or 1,700 kilowatts, during the disablement of the reciprocating engines.

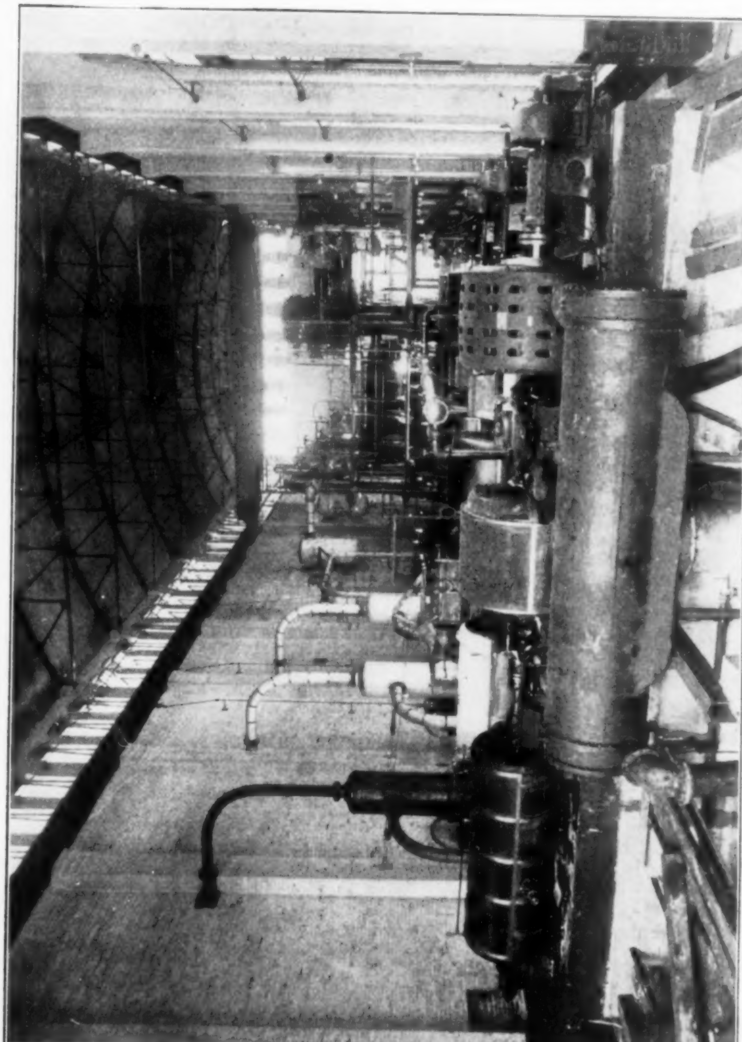


FIG. 382.—TURBINE INSTALLATION OF THREE 1,000-K. W. STEAM TURBINES AT WEST PENN RAILWAY AND LIGHTING CO., CONNELLSVILLE, PA.

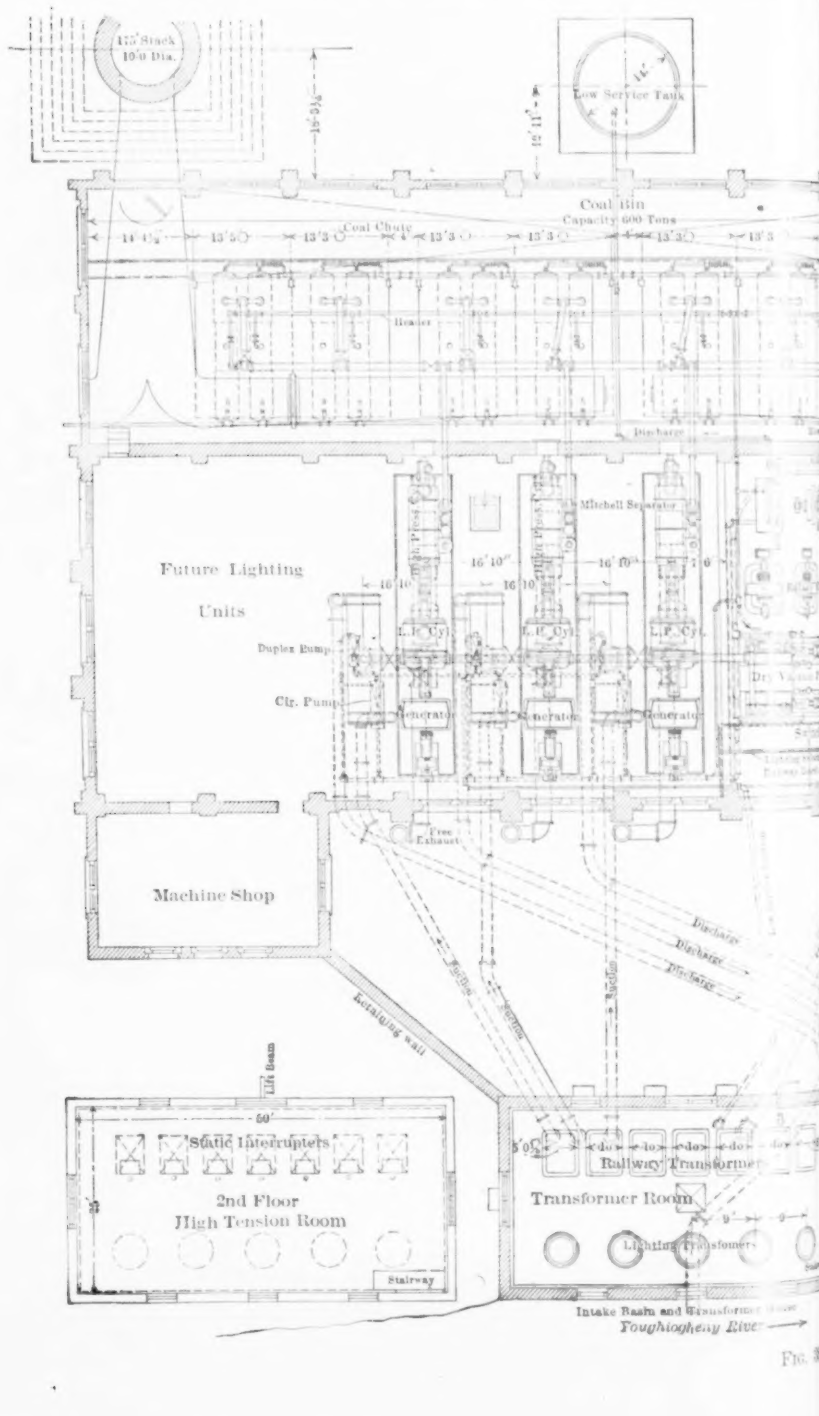
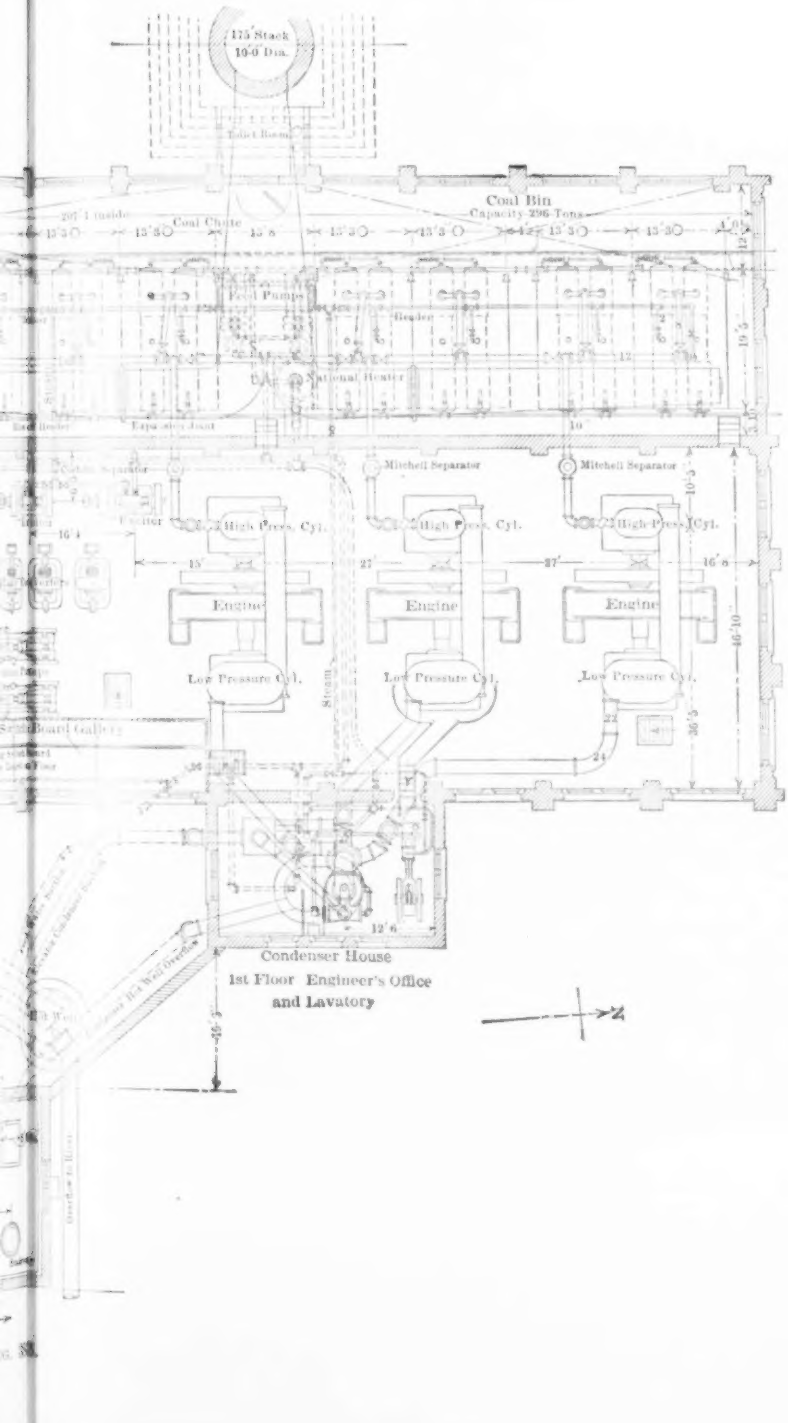

175' Stack
10'0 Dia.
Low Service Tank
Coal Bin
Capacity 600 Tons
Coal Chute
Heater
Discharge
Mitchell Separator
High Press. Cyl.
L.P. Cyl.
Generator
Future Lighting
Units
Duplex Pump
Cir. Pump
Dry Vacuum Pumps
Free Exhaust
Machine Shop
Discharge
Discharge
Discharge
Suction
Suction
Suction
Retaining wall
Lift Beam
Static Interrupters
2nd Floor
High Tension Room
Stairway
Railway Transformers
Transformer Room
Lighting Transfomers
Intake Basin and Transformer House
Youghiogheny River

FIG. 8


175' Stack
10-0 Dia.
Toilet Room
Coal Bin
Capacity 296 Tons
Coal Chute
Feed Pumps
Header
National Heater
Expansion Joint
Mitchell Separator
Mitchell Separator
High Press. Cyl.
High Press. Cyl.
High Press. Cyl.
Exciter
Engine
Engine
Engine
Low Pressure Cyl.
Low Pressure Cyl.
Low Pressure Cyl.
Steam
Switch Board Gallery
Condenser House
1st Floor Engineer's Office
and Lavatory
N

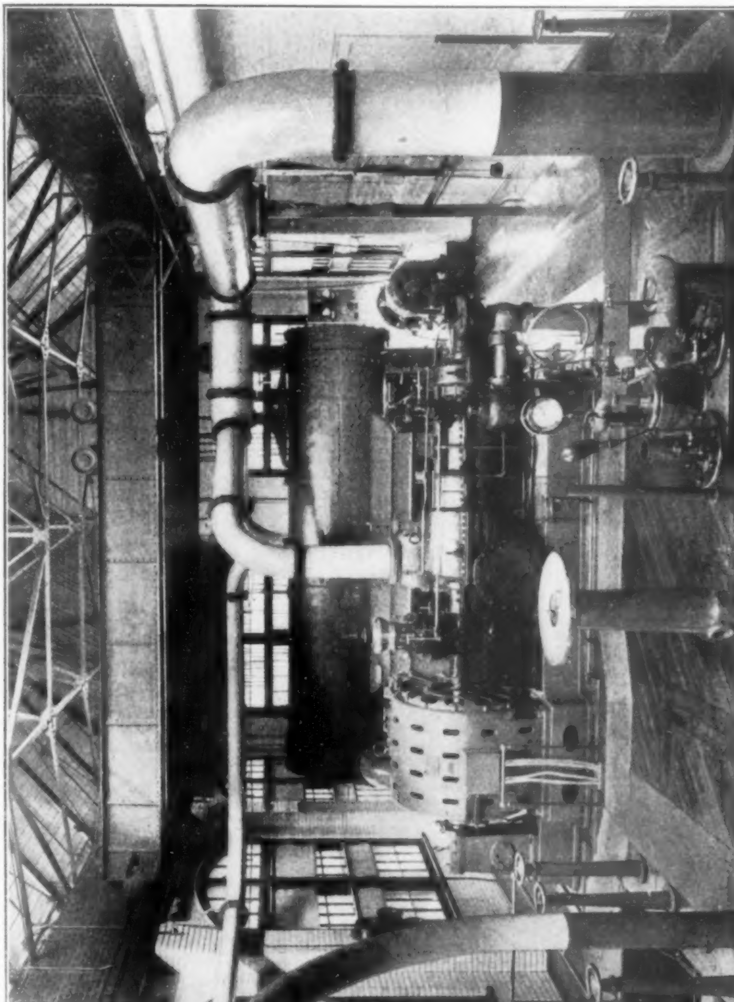


Fig. 384.—400-K. W. Installation at Yale & Towne Mfg. Co., Stamford, Conn.

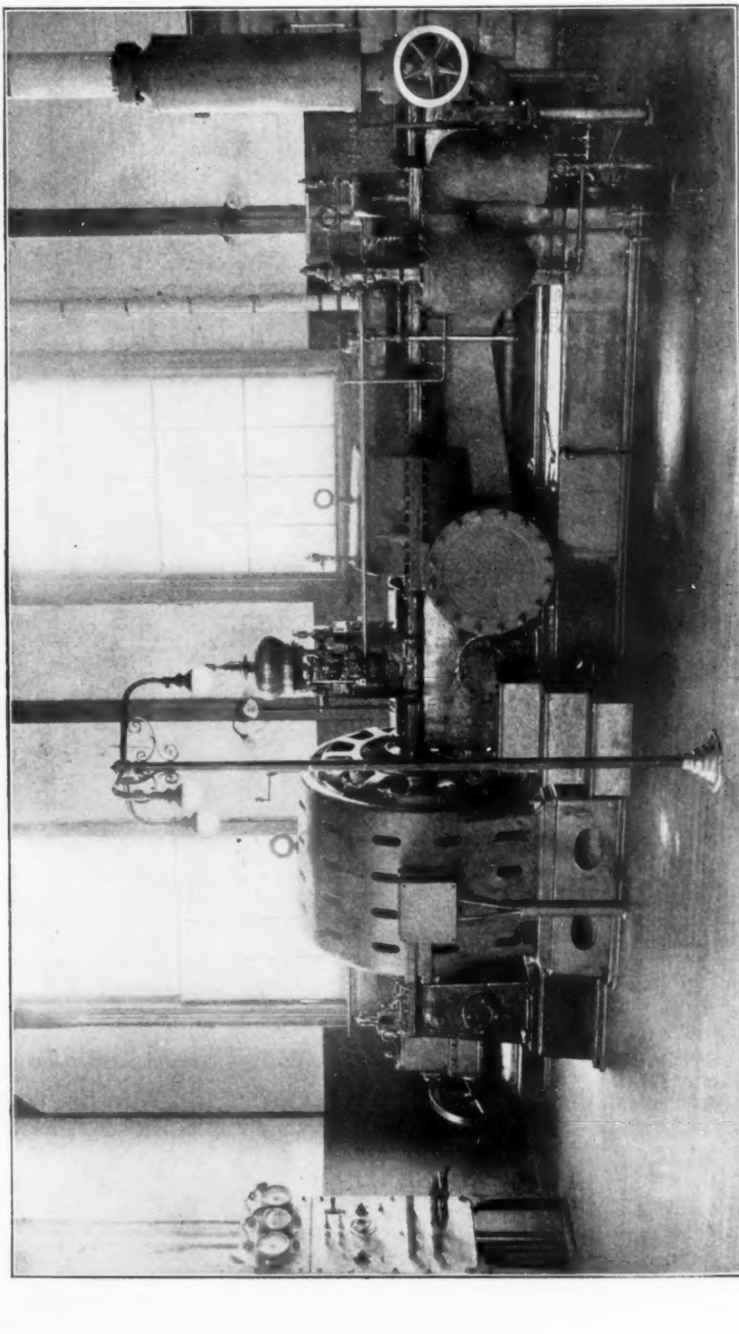


FIG 385.—400 K. W. WESTINGHOUSE-PARSONS STEAM TURBINE AT JOHNSTON HARVESTER CO.'S PLANT, BATAVIA, N. Y.

95. A plan of the power house, Figs. 380 and 381, shows the relative amount of room occupied by the turbines and reciprocating engines. The condensers and foundations are shown in Fig. 381, the condensers being placed immediately below the turbines, and the outer edges of the turbine bedplate being supported on narrow concrete walls.

96. Fig. 382 shows the power plant of the West Penn Railway & Lighting Company at Connellsville with three 1,000 kilowatt units now in operation. Reciprocating engines of similar capacity are shown in the background.

A plan of this station is shown in Fig. 383. Here, again, the space occupied by the turbines, in comparison with reciprocating engines of similar power, is exemplified.

97. A 400 kilowatt turbine plant, installed at the Yale & Towne Manufacturing Company, is shown in Fig. 384. This particular plant, which has since been increased by the addition of a turbine of equal capacity, was the subject of Mr. Waldron's paper in June, 1903.

Fig. 385 shows a 400 kilowatt turbine installed at the Johnston Harvester Company.

98. Many other installations might be mentioned, which would only serve to exemplify further such features of the steam turbine as have already formed the basis of this paper.

Suffice it to say that at present there are in operation and in course of erection in this country, in sizes ranging from 400 to 2,000 kilowatts, 43 turbines of the Westinghouse-Parsons type, the total capacity of which approximates 27,000 kilowatts.

There are also under construction at East Pittsburg, in various sizes to a maximum capacity of 5,500 kilowatts, turbines aggregating 69,400 kilowatts.

The output of one builder, including machines in operation, reaches a total of 111 turbines, aggregating 96,400 kilowatts; or an average, per unit, of 868 kilowatts.

The foregoing record is convincing that the application of the steam turbine to general power work is permanently established.

DISCUSSION.

Mr. F. A. Waldron.—Mr. Hodgkinson's paper refers to the desirability of very high vacua. This is exceedingly desirable, in so far as the water rate per hour is concerned, for the turbine

itself. The point, however, to be considered by the plant operator is, does it pay, as a whole, to run at an abnormally high vacuum.

With the apparatus at my disposal, I have been able to obtain as high as 29¼ inches of vacuum referred to barometer. The cost, however, of maintaining this vacuum does not compensate for the additional economy obtainable in the turbine. This is especially true where surface condensers are used and the water of condensation is returned to the boilers.

After a number of complete tests, it is doubtful if, in a majority of cases, where surface condensers are used it is expedient to attempt vacua above 28 inches. Undoubtedly, there are conditions that might warrant carrying higher vacua than this, but it is a matter which should be carefully considered and looked into by all who are running (or contemplate running) their plant with a turbo-generator.

I had occasion, last June, to refer in a casual way to superheat in turbine exhaust. Since presenting that paper, the condensing apparatus and arrangement of piping have been entirely changed. I have made several tests relative to superheat in the turbine exhaust, and have found that, under proper conditions of piping and condensing apparatus, the temperature in the exhaust of the turbine does not vary more than one or two degrees either way from the theoretical temperature of the vacua, with loads varying from .75 to 1.25 of the nominal capacity of generator.

Mr. E. Meden.—In paragraph 2, Mr. Hodgkinson, in speaking of the ideal turbine element, selects the Pelton wheel as representing the ideal, "because it may be capable of giving a complete reversal to the *jet*, so that the spent fluid may *issue from the buckets without any velocity.*"

I do not think a reversal of the *jet* represents the ideal condiions, although a complete reversal of its relative velocity may be necessary in Mr. Hodgkinson's ideal turbine. However, it is difficult to comprehend a fluid issuing without any velocity, and hence we must even in the ideal case assume an angle, however small, between the discharging edge of the bucket and the place of rotation of the wheel. Further, a tangential jet is only possible if a single stationary element of the elementary turbine of the Pelton type is considered, whereas we here deal with the continual process of ideal energy transmission. This will bring

other types of turbine wheels within the scope of ideal representation.

In paragraph 4, Mr. Hodgkinson speaks of the difficulty of obtaining most efficient velocity of turbine wheels on account of limitations of strength of materials. While there undoubtedly is a limit to the strength of materials, there is, however, no limit to the strength of structures of such materials if properly designed. That the turbine wheel is no exception to this rule is stated in the paper on the De Laval turbine before this meeting.

The effect of the reduced peripheral velocity in the cited case of a De Laval turbine running 32 per cent. below speed (it actually runs $27\frac{1}{2}$ per cent below the speed for maximum efficiency) is a reduction in efficiency of only $7\frac{1}{2}$ per cent. Thus, to meet a convenient design a considerable reduction in the peripheral velocity can be made without materially affecting the efficiency.

In paragraph 6, Mr. Hodgkinson states that East Pittsburgh experiments show the velocity to be higher, and the ratio between the initial pressure and the pressure in the throat of the nozzle to be lower with lower initial pressure. This is contrary to experiments made by Professor Zeuner and others.

During the Pittsburgh test was the quality of the steam observed?

In paragraphs 6 and 7, Mr. Hodgkinson speaks further of the expansion of steam in the De Laval nozzle. It is well always to bear in mind that this nozzle consists of two essential parts, *i. e.*, the converging and the diverging. The steam is expanded in both of these parts.

In paragraph 7 particularly Mr. Hodgkinson states that because the work done by the expanding steam varies directly as the square of the velocity, the volume of the steam increases much more rapidly than the velocity. I presume this relates to the diverging part of the De Laval nozzle.

In the converging part the work done also varies as the square of the velocity, but here the velocity increases more rapidly than the volume.

In paragraphs 18 and 19, Mr. Hodgkinson, in discouraging the use of high-steam velocities, seems to draw some hasty conclusions from Stodola's reference to the Levicki test of a 30 horsepower De Laval turbine.

In using Levicki's results it must not be forgotten that these

tests do not separate the skin friction of the wheel and the blower effect of the turbine wheel buckets. The blower effect is considerably more than the skin friction of the wheel body, and therefore these tests are of no value in judging the skin friction of fluid against the surface of buckets.

In paragraphs 20 and 24, Mr. Hodgkinson refers to the wearing action of steam on buckets of turbine wheels.

Mr. Hodgkinson's photographs show what steam can do to blades of Delta metal, but the conditions must have been unfavorable, as Mr. Lea has for the inspection of the members a few buckets taken out of a 300 horse-power De Laval wheel, an examination of which will show that the wear is very much less than shown in the experiments made by Mr. Hodgkinson. These buckets have been in daily operation for over 1½ years, the quality of the steam being dry saturated. The boiler pressure averaged 200 pounds, vacuum 27 inches. Absolute velocity of the steam about 4,000 feet, relative velocity about 2,800 feet.

Mr. C. V. Kerr.—Referring to Fig. 363, on page 371 of my paper on "Potential Efficiency of Prime Movers," it will be noted that the efficiency is a maximum at 21 inches vacuum. To ascertain the tendency for larger turbines and high vacuum, the efficiencies for a number of given tests of which the data are given in Mr. Hodgkinson's table of "Tests of Westinghouse-Parsons Steam Turbines" have been computed. The results are shown in the table herewith:

SIZE.	Number of Test.	Turbine.	Vacuum.	Available Heat. *B. T. U.*	Potential Efficiency.	Load ratio
400 k.w.	8	51	26.06	289.7	66.8	1.03
" "	18	55	27.01	304.2	64.8	1.00
" "	31	55	27.98	322.6	63.1	1.00
1,250 "	60	17	25.07	260.5	70.0	1.20
" "	63	17	26.05	273.3	69.4	1.21
" "	66	17	26.93	282.8	69.1	1.22
" "	83	17	27.42	300.3	67.3	1.24
			Superheat.			
400 "	31	55	0	322.6	63.1	1.00
" "	45	55	104	342.7	66.0	1.00
" "	52	55	180	366.6	67.5	1.00

The tests for effect of vacuum were made with dry saturated steam; and those for effect of superheat with a 28-inch vacuum. The range of gauge pressures for all the tests was 147.0—155.0 pounds. On the 400 kilowatts turbine the water rate is in terms

of brake horse-power; while on the 1,250 kilowatts turbine it is in electrical horse-power. For comparison, the generator efficiency is taken at 95 per cent. and the engine efficiency at 90 per cent., which combines at 85.5 per cent. The potential efficiency is, therefore, in terms of the indicated horse-power.

The results confirm previous indications that the potential efficiency decreases for the higher vacua. The chief reason, probably, is the rapid increase in heat made available by adiabatic expansion. At what point then shall we stop adding to condensing surface, volume of condensing water and dry vacuum pump capacity in order to gain another tenth of an inch vacuum? The answer must involve the whole plant as the water rate continues to decrease.

The tests for effect of superheat show an increase in efficiency for added superheat. How far the process would continue to be profitable to the plant is not answered by the data at hand. Theoretically, the increase in potential efficiency should continue indefinitely as determined by data from the entropy-temperature diagram.

So far, however, the tests examined show the rather curious contrast of an increasing potential efficiency as superheat is added as compared with the decrease in potential efficiency as the vacuum is increased.

Prof. Storm Bull.—I desire to say a few words about the steam turbine papers only, and do not see any good reason why the paper of Mr. Kerr should be mixed up with the others. I have been very much interested in these papers read by the representatives of the various steam turbines. The titles are, however, as you will see at a glance, somewhat misleading; "The Steam Turbine in Modern Engineering," "Different Applications of Steam Turbines," "The De Laval Steam Turbine," and finally "Some Theoretical and Practical Considerations in Steam Turbine Work," from which titles one would get the impression that the papers were of a general nature. The fact though remains that these papers refer to the Curtis, the Rateau, the De Laval, and the Parsons turbines almost exclusively, one for each paper. I would have very much liked if all of these men had sailed under their true colors. It is perfectly natural that the representatives of these various turbines should plead for their own machine in a paper which they had been invited to write. But in view of this they will certainly pardon me when I state that the results as

to efficiency quoted in these papers and obtained by tests conducted by the manufacturers themselves will not be recognized as facts by the engineering profession. Just as they do in athletics—I am not an athlete but I belong to a university in which I hear a great deal of athletics—I know that no record stands until it has been achieved in competition under disinterested management.

Taking up Mr. Hodgkinson's paper in particular, I will say that part of what I had intended to say has been anticipated by the written discussion read by the secretary. One of the criticisms which I desire to make is with respect to the word impact, where evidently the word impulse is meant. I may be very dogmatic, but hope that I shall get some sympathy from the secretary in the position I intend to take. On page 728 Mr. Hodgkinson speaks about " the original impact element has been made use of." Now, I have been taught almost from childhood that the very first condition for high efficiency of any kind of a turbine whether for water or steam, is that there shall be no impact. Impulse, if you please, but no impact. On page 730 I find " . . . is due to impact of steam," and on page 726 mention is made of " the impact element." I suppose that the use of the word impact in this manner on the part of the author is due to a slip, but for the sake of being on the right side I call attention to it.

Evidently Mr. Hodgkinson seems to think that the expansion of steam in a diverging nozzle is a very simple matter and is now clearly understood. I think he is very much mistaken. To judge from numerous articles in Continental European technical papers our brethren over there are very far from seeing clearly in this subject, hence the numerous experiments which have been carried on during the last few years and are being carried on now. Mr. Hodgkinson quotes some experiments by Mr. Levicky from Dr. Stodola's book " Die Dampfturbinen," but he does not say that Dr. Stodola has made a very large number of experiments on the flow of steam through nozzles quoted in the same book, that he does not offer rational explanations for various phenomena. The matter is certainly not yet clear although good progress has been made.

Why should the Pelton wheel be assumed to be the ideal one, as Mr. Hodgkinson does. If simplicity is the sole criterion which determines this, then of course the Pelton wheel or similar wheels would be the ideal ones. But, as everybody knows, the Pelton wheel is one of the late comers among turbines. We did have,

of course, centuries ago the impact wheel, which is a very different thing from the Pelton wheel, and this impact wheel was followed by the reaction wheel, the impulse wheel coming a good deal later. The description and comparison of the various turbines contained in Mr. Hodgkinson's paper is based upon this ideal case, and therefore seems of very little value. His criticisms of the turbines built by the competitors seem also very much out of place. It may be true enough that they have some of the defects found by Mr. Hodgkinson but the defects of the Parson turbine are left out altogether, although disinterested parties do not have to look very far to find as many defects in this turbine as in the others. It would have been better according to my opinion, if the first third of the paper had been left out altogether.

Mr. George I. Rockwood.—About ten or a dozen years ago those of us who belong to the American Society of Mechanical Engineers, or who did belong to it then, will remember the earnest discussions which we used to have in New York over the theory of the steam-engine. In memory, I can see the late Dr. Charles E. Emery advancing portentous thoughts; and I recall the wonderful papers of Dr. Thurston and the subtle dissent from all his conclusions by Professor Denton; and following him, the general joining of everybody in the room in the discussion.

In the last few years we have rather lacked the subject of the steam-engine as a matter of general discussion and have taken the minor steam-engine improvements made within the past few years in a more phlegmatic spirit. I have had hopes that the appearance of the steam-turbine would rehabilitate the ardor of the members and precipitate what Professor Bull deprecates in his discussion, a rabid trade discussion of the subject; for it appears that at present only those engaged in the manufacture of turbines really know anything about them.

And we stand here all of us ready to buy turbine engines in preference to reciprocating engines if we can see our way clear to do it. The question whether it is at this present time a wise venture to buy a steam-turbine is probably the most serious one before us as a society today. Personally I confess to having gone over root and branch to the newcomer; for while I know very well that the steam-engine is slightly more economical of steam—under the most favorable conditions—than the steam turbine has yet shown itself to be at full load, I believe that the

efficiency of the turbine, shown in dollars expended for horse-power returned, is higher than that of the engine. I am about to start up a turbine of my own with which to gain experience.

I do not know enough about the subject to be able to add any venom to this discussion, although I should be very well pleased to do so if I could. But I would like to argue a little about the condenser part of the problem. I was very much relieved to hear in Mr. Waldron's discussion that he had not found it essential from a financial standpoint to go into 29, 30, 31 and 32 inches of vacuum—28 inches was good enough for him. If that is so, of course it would follow that it would be well to consider whether the surface condenser is as desirable an investment as Mr. Hodgkinson has argued. We learn from his data that there is a gain of three or four per cent. when you increase the vacuum by one inch. Whether a gain of three or four per cent. on your total coal consumption will pay for the trouble of running a double cylinder dry-air pump and the cost of buying a surface condenser is a question, and I very much doubt that it will. In the case of my turbine, which is of 700 to 800 horse-power, I have paid for the condenser installation, not to exceed $650 for the entire installation—not including the time of one man for two weeks to erect it—as against several thousand dollars, the best price I could get on any type of surface condenser and dry-air pump arrangement. The vacuum guaranteed is 28 inches. I speak of the injector type of condenser, of course. I believe the injector condenser is more desirable for the ordinary case, especially for mill work, than the surface condenser, on account of its cheapness. The turbine is peculiarly satisfactory in combination with the injector condenser because it does not admit air to itself in operation; it has no valve-stems or piston rods to leak air into the vacuum. Under these conditions a Wheelock, a Bulkley, or a Schutte condenser will any one of them give very nearly all the vacuum there is if put up without leaks in the connecting pipe and joints.

I remember that nearly eighteen years ago Mr. Jerome Wheelock impressed it upon me—in opposition to my theoretical training—that it was desirable to have a very large exhaust pipe as compared with current practice, where you were exhausting to the atmosphere; and, on the contrary, you might get along with a much smaller exhaust pipe than was customary if exhausting to a condenser. And Mr. Wheelock, when he built the Chicago City

Railway engines, provided very large atmospheric exhaust pipes —pipes that had an area between three and four times the area through the exhaust valve; and his engine, it will be remembered, was always the most economical of its day, so long as he lived.

I have had some experience with the Bulkley condenser, as applied to Wheelock engines. In one case the low-pressure cylinder would naturally have a 12-inch exhaust pipe and condenser. It was to supplant an old engine where there had been an 8-inch exhaust pipe and condenser. The exhaust pipe was a very long one —not less than 60 feet in length. I left this 8-inch pipe and condenser in place just to prove to the owner that it would not do, and I found to my surprise that I had 13 pounds of vacuum realized inside the low-pressure cylinder with that 8-inch exhaust pipe on a 225 horse-power Wheelock engine. I rigged up a mercury column, connecting it at the condenser and at intervals along the exhaust pipe, and finally into the bottom of the exhaust chest, and the greatest difference registered between the vacuum at the condenser and at the chest was not over one-eighth of an inch. Again, I built an engine several years ago which had rather a small exhaust valve. It was a high-speed engine that would give from two to two and a half pounds back pressure upon exhausting into the atmosphere, while it would give a pressure in the cylinder when exhausting into the condenser that was hardly different from that in the condenser itself.

For these reasons, or rather experiences, I believe that the modern turbine theory is wrong where it claims it is necessary to have these relatively enormous exhaust pipes if a low vacuum is to be realized in the turbine itself. For example, the Curtis turbine is seen to open out at the bottom of the casing with a connection as big as the top of the condenser to which it is attached. I have on my turbine an inlet pipe only 5 inches diameter, but an exhaust pipe opening with a diameter of 20 inches. Mr. Bulkley tells me a 12-inch exhaust pipe should be large enough, and I have actually provided a 16-inch pipe.

In other words, I believe there is some element of the resistance to steam passing into a vacuum that is not covered by the mere consideration of the rapidity of flow of the steam over the surface of the pipe. Although it is true that if you carry the steam vacuum down from a medium to a very low vacuum the volume of the steam will very rapidly increase, my point is that we know by practical experience that the friction of the steam in passing

through the exhaust pipe is not increased in the same direct proportion.

Professor Jacobus.—There is one thing which seems remarkable on examining the four papers on the Steam Turbine. This is that if we take the figures given for the water consumption of the De Laval, Rateau and Curtis wheels and compare them with the results obtained for corresponding powers and pressures in the tests of the Westinghouse-Parsons turbine we will find that they are practically the same. Of course, there are differences, but these differences are within what might be expected between the tests of various wheels of the same make. It would, therefore, seem unwise to trust too much to any refined theory in predicting which of the wheels should be the most efficient, and it would also appear that Professor Rateau is on dangerous ground when he says that one class of wheel is at least 20 per cent. lower in efficiency than another. Where the figures for economy are as close as those given in the papers it is best to decide, which is the best wheel for a given line of work by noting the results obtained in practice and not to lean too much on theory.

Mr. Henry L. Doherty.—I wish to urge the speakers to keep to the subject of Steam Turbines. I think there are many men in the hall who are interested in hearing the broad ground of Turbines discussed. I have just arrived here after a twenty-four hour trip from New York simply to hear these papers discussed.

I would like to say regarding Mr. Kerr's paper that I regret to see a tendency to change from our old efficiency of rating. I am afraid that would tie us to a steam turbine or a steam engine, and we need a rating that applies to all sorts of prime movers. The internal combustion engine I believe would soon be on a basis where, considering the available heat that it might use, it would have an efficiency of over 100 per cent. You will have to have one basis for that and another basis for the steam engine.

Another thing I was anxious to see was a classification of steam turbines along rational lines. As I see it, there are perhaps two types of steam turbines; the others are mixtures of those two general types. The problem that we are facing is, first, the transformation of the heat of combustion to potential energy. Then the transformation of that potential energy to kinetic energy of fluid velocity, and then transform the latter, which I will term kinetic energy, to dynamic energy. In the De Laval

turbine the process goes on with three distinct steps, while in the Parsons turbine it goes on simultaneously.

I think there are many here who are very anxious to know about the broad features of the steam engine without regard to the manufacturer's interests, and especially the relation of the steam turbine to the other prime movers.

H. H. Suplee.—I wish to make a brief remark in regard to what Professor Bull stated. Professor Bull spoke about the fact that we do not begin to know it all about steam jets. Last summer I had the privilege of discussing this subject somewhat with Professor Stodola in his laboratory at Zurich. He had already published his book—which I believe is the best book that has been written on the subject—and he told me that he was still conducting a large number of experiments on the behavior of steam jets with diverging and converging nozzles, and that the things he found out were surprising him very much indeed. There is no doubt that there are many eddies and swirls in steam, just as we may see them in a stream of water, only we cannot see them in the steam; and Professor Stodola told me that he felt with all his experience that it was altogether too early to formulate any definite theory about the behavior of steam jets, and that he was not exactly groping, but he was working to find out additional data. I think it would be a great mistake, in view of his work, to assume that we know it all about steam jets.

Mr. George W. Colles.—There are a great many interesting questions in regard to steam turbines, and almost everybody seems to be interested in them. A great many people seem to be building them, besides the four types that we have heard discussed, although those are the only types that are on the market now. I am not an expert in steam turbines, but I want to bring forward a few points that have occurred to me, and then have the experts answer them if they will. The first is friction in steam turbines. It has been said in one of the papers that the impulse type of turbine, wherein the steam was expanded completely in one nozzle, had the great objection that the high velocity produced much friction against the sides of the nozzles and thereby detracted much from the efficiency. The De Laval people, of course, may retort against the Parsons-Westinghouse form of turbine that in that form, although they have not the high velocity, they have a very great amount of surface exposed to the steam, which amounts to the same thing so far as reducing the efficiency is concerned.

But the main question is this: What becomes of the work that is lost in friction? In general we say it is dissipated in heat, but in this case I cannot see where that heat goes unless it goes into the steam which is passing through the turbine, to be reconverted, in part at least, into work. It cannot therefore be considered as a dead loss. If the steam is superheated at the start to such a degree that, given a perfect adiabatic expansion without friction, it reaches the condenser without condensation, then with the friction its temperature will be so raised that it will be higher at the condenser, and therefore some heat will be lost. But even in that case all the heat of the friction will not be lost, because we are not expanding adiabatically, but more nearly isothermally, and an isothermal curve gives a greater amount of work than an adiabatic curve for a given degree of expansion. If, however, the expansion begins from a condition of saturation and is adiabatic, there is a considerable amount of condensation which will be wholly or partly prevented by the rise of temperature caused by internal friction. In this case, therefore, the internal friction acts precisely like superheating, to increase, not diminish, the real efficiency of the machine; with this difference, perhaps, that the friction method involves no inconveniently high temperature or special apparatus. Hence, it would seem that it matters little whether we expand in one or several stages so far as the friction loss is concerned. In fact, this seems to me an argument rather in favor of the multiple-stage-expansion turbine, because here the only place where a sensible quantity of friction-heat can be wasted is in the last of several stages, and the percentage of lost heat is diminished correspondingly.

Another important question relates to the erosion of the buckets or vanes referred to by Mr. Lea in paragraph 12 of his paper. Mr. Lea suggests a number of more or less indirect causes for this wear: chemical action, oxidation, abrasion by solid particles in the steam, and electricity; but why need we search so remotely for what lies very much nearer at hand? I saw to-day a very common process, a cold steel plate cutting through a cold steel rail, which latter was so hard that a file and cold-chisel could hardly touch it; yet the plate went through it in a very few seconds. The plate was presumably not sensibly harder than the rail, and it had no teeth, sharp edges or grit to aid it. In short, it was a simple process of abrasion of a harder metal by a softer, or a metal-abrasion by frictional contact, aided by heat. Now I cannot see

in what respect the abrasion of metal by steam at high temperatures and velocities differs from these, more especially in the light of the molecular theory of physics and the many demonstrations we have of it. Taking another example, a current of air or of the same steam will, as we know it will, evaporate water or mercury in contact with it, or, in other words, carry away particles of the surface with which it comes in contact; why should it not evaporate in a like manner hard metal in contact with it, though, of course, requiring a much greater temperature and velocity to make itself felt. If this is the fact, there are two important deductions, namely, first, that it is useless to seek to prevent the erosion by purification of the steam or otherwise, and second, that the erosion can be prevented only by using lower temperatures and velocities, which again is a point in favor of the multiple-expansion turbine, as opposed to the De Laval type.

A confirmation of this theory as to the cause of erosion lies in the facts, first, that the greatest erosion seems to take place in the De Laval turbine (where all the work is performed in a single expansion), and second, that the wear (as stated by Mr. Lea) affects only the steam inlet side of the buckets, where, as a matter of fact, the velocity is greatest with respect to the wheel.

There is one thing that the De Laval people seem to have overlooked in discussing Mr. Hodgkinsons paper. In paragraph 25 of this paper it is stated substantially that the ideal turbine (meaning the De Laval turbine) is essentially a refined form of that built by Branca in 1629. This seems to be hitting the De Laval turbine pretty hard. Some years ago when it first came out it was hailed as an original and ingenious invention. Now, I have no doubt that Dr. De Laval did invent something, but as a matter of curiosity we would probably all be interested to hear from the De Laval people as to what that something was. This is, besides, not simply a matter of curiosity, as there are a good many people who would like to know what the De Laval patents stand for, and how much ingenuity they must exert to get around them.

Mr. Alex. Dow.—Speaking from the point of view of the man who buys turbines I have to say that I buy them for the same reason that their makers manufacture them, namely, because I thereby expect to make money. I expect to buy more turbines. I believe there are more results to be had for a dollar in that form than in the form of the reciprocating engine. From the same standpoint I have to say that I think the turbine is now

suffering seriously from its friends. It is being afflicted with too much theory and too many straps. It is a little trying to a man who believes in turbines to be told that in order to get the work out of them he must install a condensing equipment which involves more study and possibly more expense than the turbine itself. I question that statement; in my own practice I am not installing such equipment. The theory of the high vacuum is correct, but there is not enough money in it.

Coming down to the practical question as to what is necessary to run a turbine successfully, I believe first of all you must have a good turbine. There are, at least, three types on the market in this country that are good machines. The question as to whether one of them is more economical by a pound of steam than the other, is a question which will stand for an infinite amount of argument here but must be settled by each purchaser on financial considerations. The triple reciprocating engine is more economical than the cheaper single non-condensing engine, but it has not yet run the latter out of the market. There is a market for both types. There will be a market for several types of turbine and for several economies.

In my business, which is the electric business, reliability is all-important. We have in our turbine equipments up to the present time sacrificed a great deal to reliability. The complication of a condenser equipment with a so-called dry air pump, is one that we may recommend to our Board of Directors on the ground of reliability thereby secured. But in recommending it we say under our breaths that it will merely let the operating engineer be somewhat lazy in looking for leaks. I don't think it is needed under normal conditions. I am not putting in a dry air pump equipment. In my official opinion the dry air pump is, as I have said, a concession to the demand for reliability of service. Unofficially, and as an engineer, I look upon it as a concession to the slackness of the operating man and to the inefficiency of the ordinary air pump. We are inclined in these latter days to forget that an air pump should be an air pump. We sometimes think that its chief function is to remove the water of condensation and the demand of the turbine engineers, for a high vacuum should recall to our memories that the name " air pump " should mean exactly what it says.

I believe the turbine will continue to be successful in service. I believe that we will continue to buy turbines, and that we will install them with comparatively simple condensing equipments.

I believe we will presently rediscover the fact that we can get satisfactory commercial results in a condensing plant by making all the everyday points in the game of good steam practice. I do not believe that in all cases we need big condensers. I am sure that in many cases the ejector condenser, as recommended to-night by Mr. Rockwood, will answer the purpose. There is no question that the ejector condenser with a sufficient supply of water is reliable, and it is certainly cheap. It is not so well known and appreciated in the United States as it should be. In many cases the syphon condenser is desirable, but in other cases it has not been entirely acceptable, because of the fact that it depends for its comparative efficiency upon the utilization of the vacuum for lifting the water. Any disturbance of the electric service tends to a sudden disturbance of that vacuum which may initiate a sequence of troubles ending in the entire loss of vacuum and interruption of service. In practice it has been found necessary in certain classes of electric work to run the circulating pumps for syphon condensers at the speed which will deliver sufficient water without the assistance of the vacuum. This would not be necessary in ordinary commercial work. In other cases the dry air pump has been added to the syphon condenser, especially to obviate this possibility of lost vacuum due to a sequence of disturbances initiated in the electrical system.

I think that in turbine practice in the future we will most frequently use a surface condenser, and in many cases will necessarily do so, and that we will be able to obtain the desirable extra inch of two inches of vacuum without the cost and trouble of a complex installation. First, we will use feed water which is free from entrained air or gases in solution. Feed water, particularly when it is taken from the ærated discharge of a syphon or jet condenser, contains much air and other non-condensible gas. If it is sent to the boiler through a closed system of piping the non-condensible gas must escape through the turbine, and be accounted for by the condenser and air pump. If on the contrary the feed water is simmered or boiled in an open heater much of this non-condensible gas can be got rid of. Second, we require tight piping. As I have said before we far too often see the dry air pump making good the negligence in the care of the piping system. Third and last, we must use air pumps with small clearance, and of a design that does not mix the vapor with the water. These three things together will in most instances secure without

difficulty the 26 or 27 inches, which represents good condenser practice, and with a reasonable increase in condenser surface will secure the 28 inches that the turbine engineers ask for, and which certainly is desirable with turbines.

I believe in the turbine, gentlemen, for the causes I have assigned. There is money in it for the man that has to run it. I do not think it is yet in its final form. I think that all three types of turbine described to-night will go on together, and that out of them our future practice, certainly in electrical work, will arrive at a standard.

*Mr. Francis Hodgkinson.**—I feel some justification in the selection of the Pelton type of wheel for the ideal turbine element, firstly, because it consists of a single bucket element and a single nozzle element, and secondly, it more nearly approaches permitting a complete reversal of the operating fluid, and if this is attained, the spent fluid has no component parallel with the axis as is generally the case with all steam turbines.

With regard to Mr. Meden's remarks on the strength of revolving wheels, I think that while the strength of structures of material may be considerably less limited than the strength of the materials themselves, nevertheless, the strength of the structures must certainly have some limit, and I think most engineers would be considerably exercised if they were called upon to design a wheel, say 10 feet in diameter, to have a peripheral speed of say 2,000 feet per second.

In reply to Mr. Meden's question on the subject of the experiments with nozzles, referred to in paragraph 6 of the paper, would say that the steam was maintained as closely as possible "dry saturated," by having a receiver of relatively large capacity immediately in the rear of the nozzle, before entering which the steam was throttled to give the desired initial pressure for the nozzles.

A small quantity of water was allowed to remain in the receiver, which would be picked up, supposing the steam to be superheated by being wire drawn through the valve.

I rather think Prof. Storm Bull is right in his criticism of the words "impulse" and "impact." However, these two words do not appear to me to particularly well describe either of the conditions. We say, for instance, that a gas engine receives an

---

* Author's closure, under the Rules.

"impulse" every so many revolutions, from which it is inferred that the action is not continuous and that work is done in "impulses," which seems to me is an entirely different sense to what is intended in the properly constructed turbine bucket. Nevertheless, I think Professor Bull is right, and I would have been glad to change the text of my paper in this respect.

Regarding my discussion on steam jets, I did not intend to convey the idea that their behavior was understood. As has been pointed out by two members in discussion, there are all kinds of eddies and swirls that render observations most confusing. My object was to show the general underlying principles made use of in the design of nozzles, and to show by means of a theoretical design the reason for the divergence which is frequently not understood by engineers not familiar with turbines.

Of course, we do not know very much about skin friction in turbines. Its effect, as pointed out by Mr. Colles, must result in the heat thus generated being returned to the operating fluid. If the turbine is a single stage turbine, then this heat is notwithstanding lost. If, however, it is multiple stage, then it would seem that this heat would not be entirely lost except in the last stage.

It is probable that frictional losses in turbines vary with a higher exponent than the square of the relative velocities. The number of stages in a turbine designed to work between given pressure limits, varies substantially inversely as the square of the velocity employed.

Therefore, it would seem that by increasing the number of stages in a turbine, the frictional losses would decrease more rapidly than the increase of surface.

This would seem an additional reason to favor the multi-stage type to that pointed out by Mr. Colles.

Regarding Mr. Colles's discussion, I am sorry to have conveyed the least impression of stating that Dr. De Laval had invented nothing. While, I think, I am correct in stating that the De Laval turbine is a refined form of the Branca wheel, the degree of refinement involved however, is very great.

The difficulties overcome, that have been pointed out in the paper by Messrs. Lea and Meden, bear witness to the progress made. The developments, as shown by the invention of the divergent nozzle; the construction of the disc and buckets; the employment of a flexible shaft, and the remarkable work done by the De Laval gears, demand the admiration of engineers.

No. 1038.*

DIFFERENT APPLICATIONS OF STEAM TURBINES.

BY PROFESSOR A. RATEAU, OF PARIS.

1. One of the most interesting developments in mechanical engineering during the years 1902 and 1903 was the greatly extended use of steam turbines. This development is the result of efforts which have been made by various people to increase the efficiency of turbines to such a point that they may compete commercially with reciprocating engines. The author has been occupied for more than fifteen years in the study of rotary engines, and he has already published several papers upon the subject. He believes that in presenting to the Institution of Mechanical Engineers an account of the practical result of his labors upon steam turbines he will contribute usefully to the study of this important question.

2. Although the principles which distinguish the different kinds of turbines are well known, it may be useful to recall briefly their distinctive characters in order that his own type of turbine may be more easily distinguished. In common with all other steam-engines, turbines transform into mechanical work the energy given out by steam during its expansion from the initial pressure of admission to the pressure at the exhaust. But whilst reciprocating engines effect this transformation of energy by means of variation in pressure of the steam, turbines can effect this transformation both by means of the pressure and by means of the velocity of the steam while expanding. The employment of the velocity only in each moving wheel characterizes the action or impulsion turbines, among which may be cited the Laval and Curtis turbines, as well as that designed by the author; whilst the simultaneous employment of the velocity and partial use of the pressure characterize the re-action turbine, of which the best-

---

* Presented at the Chicago meeting, May and June, 1904, of the American Society of Mechanical Engineers, and forming part of Volume XXV. of the *Transactions*.

known type is that of Parsons. Whatever may be the method in which the steam acts in the turbine the chief problem consists in the employment, with good conditions of efficiency, of the very great velocities attained by the steam in expanding. When the expansion takes place in one stage, as in turbines with a single wheel, then the velocity of flow reaches, as is well known in a condensing engine, a value which is usually above 3,600 feet per second. But in order to obtain the maximum efficiency, the moving part of the machine should have a relative velocity which is approximately the half of that of the steam. As it is practically impossible to construct turbine wheels suitable for running with a peripheral velocity above 1,200 feet per second, the efficiency of turbines with a single wheel is necessarily low, this being due chiefly to the necessity for the employment of diverging inlet nozzles, which give rise to great losses of energy by friction and eddying. On the other hand, angular velocities which correspond to these peripheral speeds prevent the direct driving of dynamos, and render necessary reduction gears of special and costly construction, which, however, cannot be protected from excessive wear, and are exposed to accidental breakage.

3. A consideration of these circumstances has induced inventors to divide the expansion of the steam into successive stages, and thus to produce turbines with multiple wheels, which are nothing but a series of simple turbines mounted upon the same shaft and driven successively by the same current of steam. This design of multiple turbines is by no means novel. It will suffice to mention the name of Tournaire, a French mining engineer, whose theoretical description to the Academy of Science in 1853 of a re-action turbine with multiple wheels is surprising when the description is compared with the Parsons turbine brought into use 30 years later.

4. Every simple turbine may be designed either as an impulse turbine or as a re-action turbine. In the former kind the fall of pressure under which the simple turbine works takes place solely in the distributor, whilst in the latter type the fall of pressure takes place, not only in the distributor, but also in the moving wheel, so that in the latter type there is a higher pressure in the guide-vanes than at the exit. Fig. 386 represents the guide-vanes and the moving vanes of an impulse turbine in which it is supposed that the fall of pressure is small enough at the vanes to render unnecessary the use of diverging nozzles as in the Laval turbine.

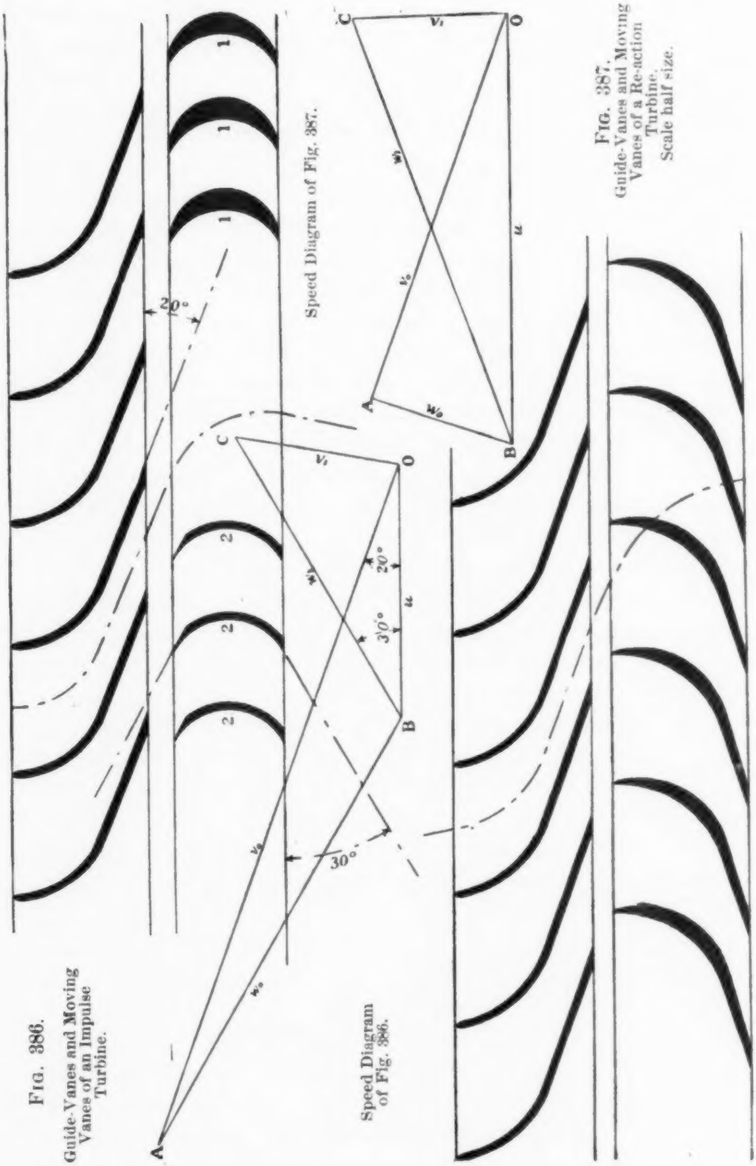


FIG. 386.
Guide-Vanes and Moving Vanes of an Impulse Turbine.

FIG. 387.
Guide-Vanes and Moving Vanes of a Re-action Turbine.
Scale half size.

In the illustration the triangles of velocities are given at the entrance OAB, and at the exit OBC of the moving wheel. Similarly in Fig. 387 are shown the guide-vanes and the moving vanes of a re-action turbine, and the triangles of velocities at the entrance OAB and at the exit OBC of the moving wheel are shown in the figure. It is obvious from these illustrations that, by the mere inspection of the moving vanes, it is possible to distinguish between an action turbine and a re-action turbine. In the former type the vanes appear in transverse section in the form of an arc of a circle with entrance and outlet angles practically the same, whilst in the re-action turbine the cross section of the vanes is in the form of a parabolic arc, the entrance angle being more or less approximately 90 degrees whilst the outlet angle is generally between 20 degrees and 30 degrees. Generally the vanes are increased in thickness in the middle, so that the steam spaces between them have approximately a constant transverse section for impulse turbines or decreasing section for re-action turbines. If however they are made of plate they may have a constant thickness as in 2, Fig. 386. In re-action turbines it is important that the increase in thickness should be suitably calculated, otherwise there will be an important loss of efficiency. It is otherwise with the impulse turbine in which the efficiency is very little affected whether the vanes have a constant or variable thickness. In a given turbine with multiple wheels the different simple turbines may be all designed in the same way or some may be designed in one way and some in another way to produce a hybrid system.

5. *Drum Turbines and Multicellular Turbines.*—From another point of view, drum turbines may be distinguished from multicellular turbines, in the former type of which the Parsons turbine is the best known example, the moving vanes are fixed upon the periphery of a cylindrical drum; in the latter they are fixed to the periphery of wheels more or less flat and separated from each other by diaphragms which divide the interior of the turbine into cells. The turbine of the author belongs to the latter type.

6. Fig. 388 is a diagram of a drum turbine, and Fig. 389 a diagram of a multicellular turbine. These two types have different properties, so far as losses of steam are concerned. In the multicellular action-turbine leakage of steam can only take place through the play c, Fig. 389 which exists between the shaft and the ring of the fixed diaphragm which surrounds this shaft. In the re-action drum turbine, on the contrary, leakage may take place: 1st, around the moving wheels in a, Fig. 388 (by reason of

the re-action); and 2d, in *b* through the play between the guide-vanes and the moving drum. The first leakage can be suppressed, if desired, by making the wheels work by impulsion instead of by re-action, but then the second leakage between the fixed vanes and

ESCAPE OF STEAM AND LONGITUDINAL THRUST IN TURBINES.

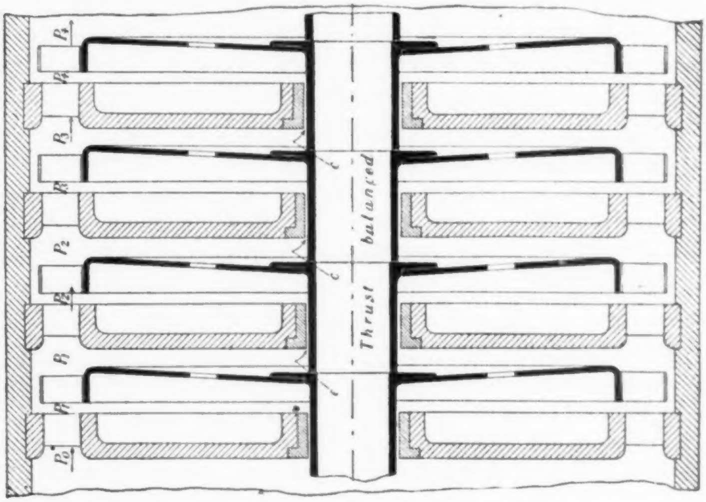


FIG. 389.—MULTICELLULAR TURBINE (Rateau.)

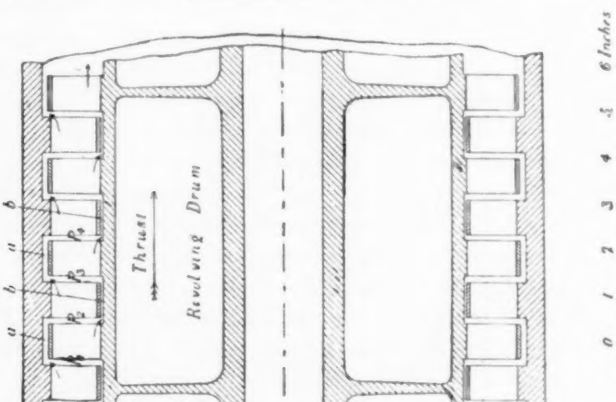


FIG. 388.—RE-ACTION DRUM TURBINE (Parsons).

the moving wheels attains its maximum. It is because of this leakage at the periphery of the drum that turbines of the Parsons type require the most accurate workmanship. It is easy to understand that a play of some tenths of a millimetre is sufficient, at least upon the high pressure side, to produce a cross-sectional area of the leakage passages equal to that of the admission passages

between the distributing vanes; if by especially exact workmanship the cross section of the leakage passages is made very small, is it not to be feared that after some years of use the wear of the parts, either by friction of the moving parts against the fixed parts or by the action of the steam, will increase the cross section of the leakage passages and so cause a notable decrease in the efficiency of the machine?

7. In multicellular turbines the leakage, being confined to the periphery of the shaft, is reduced in proportion to the radii of the shaft and the drum; and moreover, friction being less to be feared at a place where the relative velocities are smaller, the play may be reduced to the absolute minimum. In practice we do not trouble about giving this play any precise value. We build the machine with practically no play round the shaft, and when started the machine itself makes play sufficient to turn without touching the internal rings of the diaphragms.

8. *Friction of the Moving Parts on the Steam.*—It would appear probable that multicellular turbines produce relatively large losses by friction of the moving parts upon the steam owing to the large surface of the moving wheels. This is however not the case. From tests which we have made to determine the law of friction, and also from the results of experiments made upon complete machines, we have found that frictional losses represent only from 2 to 4 per cent. of the normal power developed by the machine. These figures apply to turbines with an output exceeding 500 horse-power. These figures are comparable with those obtained in drum turbines in which the friction of the moving vanes themselves must be added to that of the cylindrical surface of the drum and of the balancing pistons.

9. *Turbines with Groups of Wheels.*—Another type of turbine which may really be considered as belonging to the impulse class is being built upon a large scale, and is known under the name of Curtis. In this type instead of using upon a single moving wheel the velocity of discharge of the steam leaving the distributor, it is used upon several wheels arranged in series in order to diminish the velocity gradually. The principle upon which the Curtis turbine is designed has been described in 1890 by Mr. Mortier in a discussion which took place after a paper had been read by the author upon steam turbines in general and upon the Parsons turbine in particular.*

* See Comptes Rendus de la Société de l'Industrie Minerale à St. Etienne, Meeting of April 12, 1890.

10. The type which we have described as having a group of wheels possesses the notable advantage of allowing a great reduction in the speed of rotation for wheels of the same diameter and of the same number. It would therefore be preferred to all others if it had not, in our opinion, the grave defect of preventing a sufficiently high efficiency being attained. It is easy to see that, owing to the great velocity of flow of the steam in the first wheels, the losses of energy by friction and eddying are very great, reaching such a value that the second wheel develops less power than the first, and the third wheel and the fourth, when more than two are used, act more like brakes than prime movers. The Curtis turbine, as originally constructed at the Fisk Station in Chicago, had groups of four wheels, but the process of evolution reduced these to groups of three and then to groups of two wheels. Even with such a reduction in the number of wheels in a group the efficiency, according to our calculation, is still at least 20 per cent. lower than that obtainable with the multicellular turbine which has only one wheel per cell. The author therefore ventures to think that the Curtis type of turbine will disappear, as in the process of evolution it will become the multicellular type pure and simple.

11. *Description of the Rateau Turbine.*—Since 1894 the firm of Sautter-Harlé of Paris, with the assistance of the author, has been experimenting upon the construction of steam turbines. The first turbine constructed had only a single moving wheel, like the Laval turbines, and this wheel was formed with vanes in the form of a double arc similar to the buckets of a Pelton wheel, and these vanes were milled out of a solid block of steel. This type has since been copied in the Riedler-Stumpf type, built by the Allgemeine Elektricitäts Gesellschaft, but which was rapidly abandoned by us as it did not offer any chance of obtaining the maximum efficiency. The most recent Rateau turbine is of the action type, that is to say, expansion of the steam is fully carried out in the distributor for each group consisting of a distributor and one moving wheel. The steam therefore acts by its velocity and not by its pressure. These turbines are moreover multicellular, that is to say, they consist of a certain number of elements, each element comprising one distributor and one moving wheel. A very interesting characteristic of the type of action turbines is the possibility which it allows of leaving very considerable play between the fixed parts and the moving parts, and this greatly facilitates construction and obviates

the chances of dangerous friction if the bearings should become worn or the shaft somewhat bent. Besides this the wheels revolve in a chamber where the pressure is uniform. There is for that reason an absence of longitudinal thrust upon the moving parts and no necessity for the use of dash-pots for the purpose of overcoming the effect of this thrust, although such dash-pots are neces-

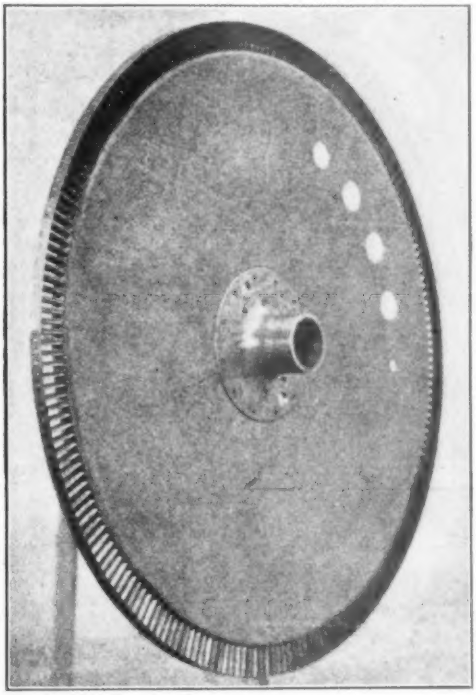


Fig. 390.

-sary in re-action or drum turbines. Finally, in the action type partial injection of steam is possible, that is to say, the steam may be directed upon a limited portion of the circumference. The moving wheels are formed of discs of sheet steel more or less thin, and upon the periphery of these discs are riveted vanes of cylindrical form, Figs. 390 and 391. A steel band riveted to the periphery maintains the correct spacing of these vanes and insures great rigidity to the construction. Wheels so constructed are extremely light, and remain in equilibrium at a velocity far higher than that

to which they are subject in the turbine. These moving wheels turn between circular diaphragms provided with distributing vanes which enter circumferentially into grooves formed in the interior of the turbine case. Between two adjoining diaphragms is therefore produced a cell or very flat chamber in which the moving wheel revolves. The shaft passes through the diaphragms in

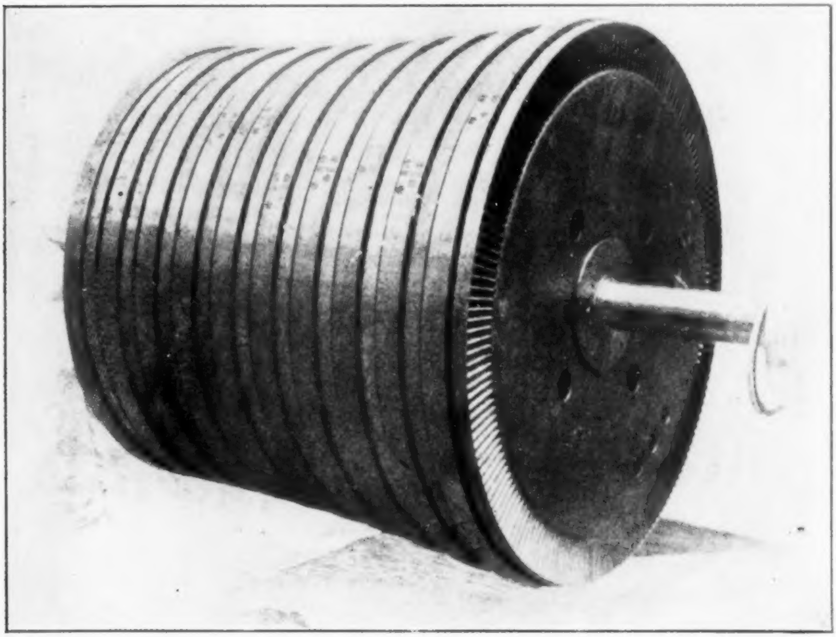


FIG. 391.

collars of anti-friction metal with very slight friction. In the first diaphragm which the steam passes through the distributing vanes are placed only upon a part of the circumference. Partial injection of the steam is therefore obtained, and thus the velocity of the steam is better utilized. Moreover, to produce the same effect the useful part of each distributor is set with an angular advance on the preceding section; this angle of advance is calculated according to the speed of rotation, so that the steam leaving one moving wheel enters into the following distributor and never encounters a solid wall which would produce a shock and therefore a loss of kinetic energy. For the last wheels it is necessary to

employ total injection, that is to say, the distributing vanes must be set upon the whole circumference of the diaphragm, and moreover, owing to the expansion of the steam, the radius must be increased. The bearings of these turbines are external, and by means of a special system of spring packing they are kept perfectly tight. No oil is carried by the exhaust steam to the condenser, and this has an important advantage as the water of condensation can be used for feeding steam boilers direct without any necessity for the use of an oil-separator. The speed of rotation is controlled by a centrifugal governor with a Denis compensator, which acts upon an obturator which controls the pressure of steam entering the turbine.

12. *High and Low Pressure Turbines.*—In the case of installations where exhaust steam must be used, a subject which we shall consider in detail at a later stage, it is often desirable to supply the turbines temporarily with steam at high pressure when the primary machine is stopped or is furnishing less steam, and the work done by the turbine must remain the same. In order to obtain economical working it is preferable not to expand this steam in order to lower it to the pressure for which the low-pressure turbine is built. Therefore, the author has designed a turbine which may be described as of the mixed type, and this can be supplied either simultaneously or separately by steam at high pressure and by steam at low pressure without any lowering of the efficiency of the mechanism. In order to attain this result the turbine is constructed in two parts, one designed for high-pressure steam and the other for low-pressure steam. The steam at high pressure having done work in the first portion will pass to the second portion, which may be fed either by steam coming from the accumulator or by the exhaust from the first portion. The admission of high-pressure steam into the first portion is automatically obtained by means of a special regulator which allows steam from the boilers to pass to the first portion as soon as the pressure in the accumulator falls below a given value. This arrangement, which has been adopted in all the new applications of the system to the use of exhaust steam, works very economically, and is particularly suitable to cases in which the primary machine works irregularly, and where therefore the demand for live steam is frequent and somewhat prolonged. In the small machines the two portions are usually joined together.

13. *Efficiency of Turbines.*—It is essential that the word effi-

ciency should be clearly defined in order to avoid any misconception as to the figures which are given below. The author uses the expression "Theoretical consumption of the perfect machine" to denote the maximum work which the steam is capable of supplying when starting from the saturated or superheated condition in which this steam is delivered to the engine and expanding adiabatically with no loss of admission pressure P to exhaust pressure p. By comparing the actual consumption of steam as measured during the tests of the machine with this theoretical consumption of the perfect machine for identical conditions of pressure and similar states of the steam the net efficiency of the machine is obtained. After special study of the question, the author has been able to draw a curve of theoretical consumption and to derive from it the following empirical formula for use when the steam is saturated and dry at admission:

$$K = 0.85 + \frac{6.95 - 0.92 \log P}{\log P - \log p}.$$

14. This formula gives the consumption K in kilograms per horse-power-hour of 75 kilogrammetres as a function of the absolute pressures P and p expressed in kilograms per square centimetre. In British measures this formula becomes—

$$K = 2.13 + \frac{16.20 - 2.05 \log P}{\log P - \log p}.$$

K in lbs. per horse-power-hour, P and p in lbs. per square inch. The curve which is given in Fig. 392 will readily afford interesting information, particularly if it be necessary to estimate the consumption of steam with a given machine to obtain a certain useful effect. In that case the theoretical consumption per horse-power-hour given by the curve should be compared to the exact consumption of steam actually observed with the machine, and of course referring to the particular number of electrical horse-power developed by the dynamo driven, or to the power in water raised (expressed in horse-power-hours), if pumps are driven by the machine. The relation between these two figures of consumption gives the combined efficiency of the group, and therefore takes account of all the losses of the steam engine and of the driven mechanism. This combined efficiency is merely the product of the efficiency of the prime mover multiplied by the efficiency of the machine driven. If the steam admitted to the turbines is superheated, then of course in estimating the theoretical consump-

tion the extra calorific energy corresponding to the superheat must be taken into account;* but for the exact estimation of this extra energy we have not as yet adequate precise knowledge of the specific heat of the vapor of water. At present we have merely the figure obtained by Regnault after very cursory experiments (0.48), a figure which is merely a rough approximation.

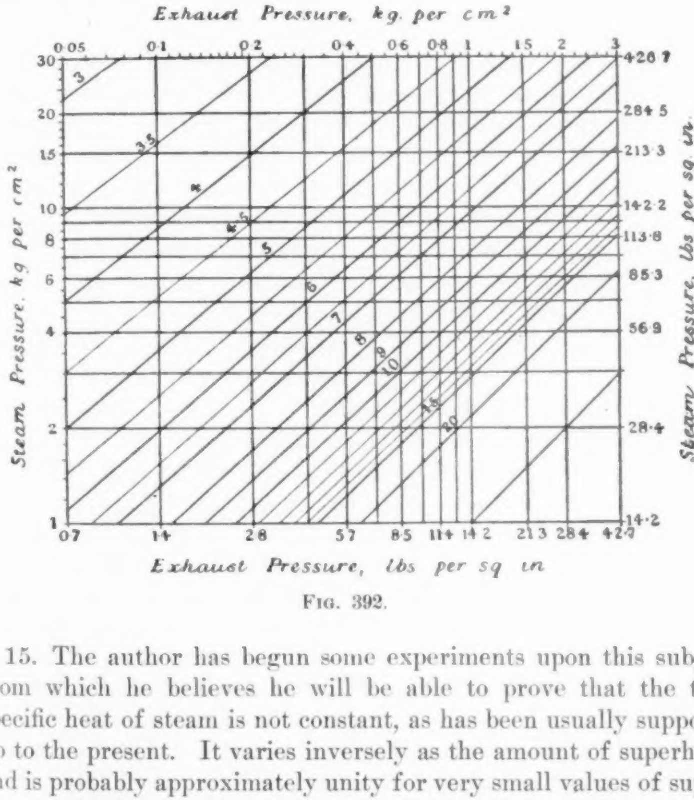


FIG. 392.

15. The author has begun some experiments upon this subject from which he believes he will be able to prove that the true specific heat of steam is not constant, as has been usually supposed up to the present. It varies inversely as the amount of superheat, and is probably approximately unity for very small values of superheat, and approaches 0.40 for large amounts of superheat.

16. *Calculation of the Efficiency of Turbines.*—The efficiency of a turbine may be calculated *a priori* when a preliminary study has been made of the practical coefficients which must be introduced into the theoretical formula. We calculate this efficiency with quite a remarkable degree of accuracy

* These results were published for the first time in the *Annales des Mines de Paris*, in February, 1897, and since that date have on many occasions been republished in various French and foreign publications.

by dividing the losses of the machine under two headings: 1st, the internal losses which are produced by friction and eddying of the steam in the fixed and moving vanes; 2d, the external losses which correspond to the leakages of steam in the play between the fixed and moving parts, and to the friction of the wheels upon the steam and to the friction of the bearings. The first kind of losses gives rise to the *internal efficiency*, which may therefore be called the "hydraulic efficiency" by extending the use of the term employed for hydraulic turbines. This internal efficiency depends upon the more or less perfect form of the vanes, and also to the relation between the peripheral speed of the moving vanes and the speed of flow of the steam. It is possible to draw a curve showing the efficiency as a function of the peripheral velocity, and therefore it is easy to at once assign the degree of efficiency which a given turbine will realize.

17. Once having fixed the internal efficiency, then, in order to obtain the net efficiency, we must deduct the losses by leakage as well as the losses by friction of the wheels upon the steam, and also those in the bearings. Let us take, for example, a multicellular turbine of 1,500 brake horse-power upon the shaft with a speed of 1,500 revolutions per minute. By means of practical coefficients found by experience we can calculate that the internal efficiency of such a machine may easily rise to 69 per cent.; on the other hand, the losses by leakage and by friction in the bearings absorb 1·5 per cent. of the normal power, and the losses due to friction of the wheels upon the steam amount to 2.5 per cent., making a total for the external losses of 4 per cent. The net efficiency upon the shaft at the speed stated will then be 0.69 multiplied by 0.96=0.66. From this value of efficiency it is easy to calculate the consumption of steam per horse-power-hour which would be required by this turbine under conditions of pressure and of superheat of the steam already determined. All calculations for designs of turbines that we have made by this means have always proved correct within 1 to 2 per cent. of actual practice. Such accurate calculations are not possible in the case of reciprocating engines owing to the action of the cylinder walls, which makes calculations uncertain and often inaccurate, whilst in the turbine the continuity of flow of the steam allows practical calculations of a high degree of accuracy to be based upon theory once having determined the fundamental coefficients which are employed in the formulæ. For a more detailed consideration of the use of these coefficients the

author refers to the work which he published in 1903 in the "Revue de Mécanique" entitled "Théorie élémentaire des Turbines à Vapeur."

18. *The Results of Actual Practice.*—The author began the construction of his first multicellular turbines in the year 1898 in collaboration with the firm of Messrs. Sautter-Harlé of Paris, and there are now already at work or in process of manufacture turbines developing more than 25,000 horse-power in units varying from 10 to 2,000 horse-power irrespective of the designs now in preparation. Some of these turbines are used for driving dynamos, others for pumps, for fans, and for the propulsion of vessels. The author has appended a brief abstract of the conditions and economical results of the most interesting of these installations.

19. *Turbo-Dynamos for Direct Current.*—The company of the mines of Peñarroya in Spain installed in their central electric lighting station a little more than a year ago three groups of turbo-dynamos developing 500 electric horse-power with direct current at 240 volts. Each of these three generating sets comprises two turbines for high and low pressure, and two dynamos for direct current, the latter directly driven from the turbines upon the same shaft and upon the same bedplate, and the two dynamos supply a three-wire network with a potential of 480 volts between the outer wires. The speed of rotation is about 2,200 revolutions per minute, and the floor space occupied by the turbine with its two portions is only 12 feet by 5 feet 6 inches, with a height of 5 feet inclusive of a bed-plate 1 foot high. The condensation of the steam is carried out by means of ejector-condensers of a type designed by the author.

20. Figs. 393 and 394 show a longitudinal section and a plan of the turbine which has twenty-four moving wheels. The works tests of the first group were made in September, 1902, with the greatest care, and they have given the results which are shown in Table I. The condensation was made for these tests by means of a surface condenser belonging to the works, so that the water of condensation might be collected and measured. It will be seen that the vacuum at the condenser declined as the power increased. This result arises from the fact that the condenser was designed for machines of 250 horse-power only, and therefore was of insufficient size for the larger volumes of steam.

21. It follows from the preceding figures that a turbine developing 644 electric horse-power and working without appreciable

superheat between an admission-pressure of 156 lbs. absolute and an exhaust-pressure of 1.8 lbs. (notably higher than the results given by good condensers in practice) has given the reduced consumption of 14.9 lbs. per electrical horse-power at the terminals; the combined efficiency for the set being then 59 per cent. when compared with the energy contained in the steam for the same fall of pressure. With steam at 180 lbs. superheated 50 degrees and a vacuum of 29 inches of mercury, the consumption at 2,400 revolutions per minute would decrease to 11.5 lbs. per electrical horse-power-hour.

TABLE I.

Data.	¼ Load.	½ Load.	Full Load.	Over-load.	Over-load at 2,400 rev.
Electrical H.P. at brushes....	135	259	525	627	641
Admission pressure, absolute, lbs. per sq. in.	46.21	76.6	136	156	156
Exhaust pressure, absolute, lbs. per sq. in.	1.24	1.33	1.63	1.82	1.82
Theoretical steam consumption of perfect engine per H.P. hour.............lbs.	10.93	9.8	8.89	8.73	8.73
Actual steam consumption per electrical H.P. hour at brushes...............lbs.	21.3	18	15.8	15.39	14.90
Combined efficiency of the electrical generating set....	0.513	0.540	0.560	0.569	0.580

22. Fig. 395 gives the curves corresponding to the figures taken at a speed of 2,400 revolutions per minute. The abscissæ denote the power of the machine in electric horse-power at the terminals of the dynamos. The ordinates vary with the curves. For curve *A* the ordinates denote the absolute pressure of the steam on reaching the turbine in lbs. per square inch. For curve *B* they show the absolute back pressure at the exhaust in lbs. per square inch. For curve *C* they show the total consumption of steam in lbs. per hour, and for curve *D* they show the consumption per electrical horse-power in lbs. per hour, and lastly for curve *E* they show the combined efficiency. These graphic curves show clearly a remarkable characteristic of turbines which is the very low relative consumption of steam at small loads. It will be seen that in order to drive the machine at its normal speed with no load but with the dynamo excited, the total consumption of steam is only 10 per cent. of that at full load. With reciprocating engines the result would be very different. Under similar conditions the

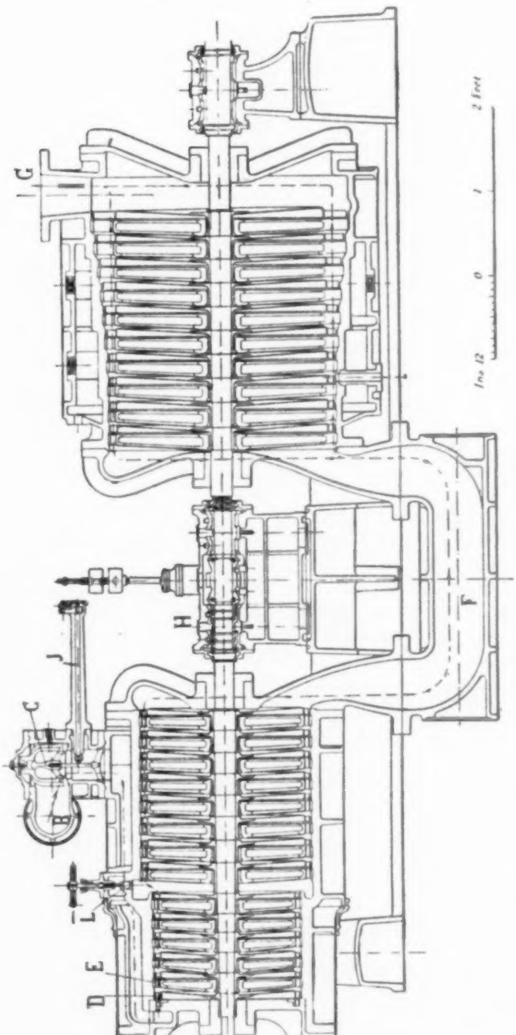
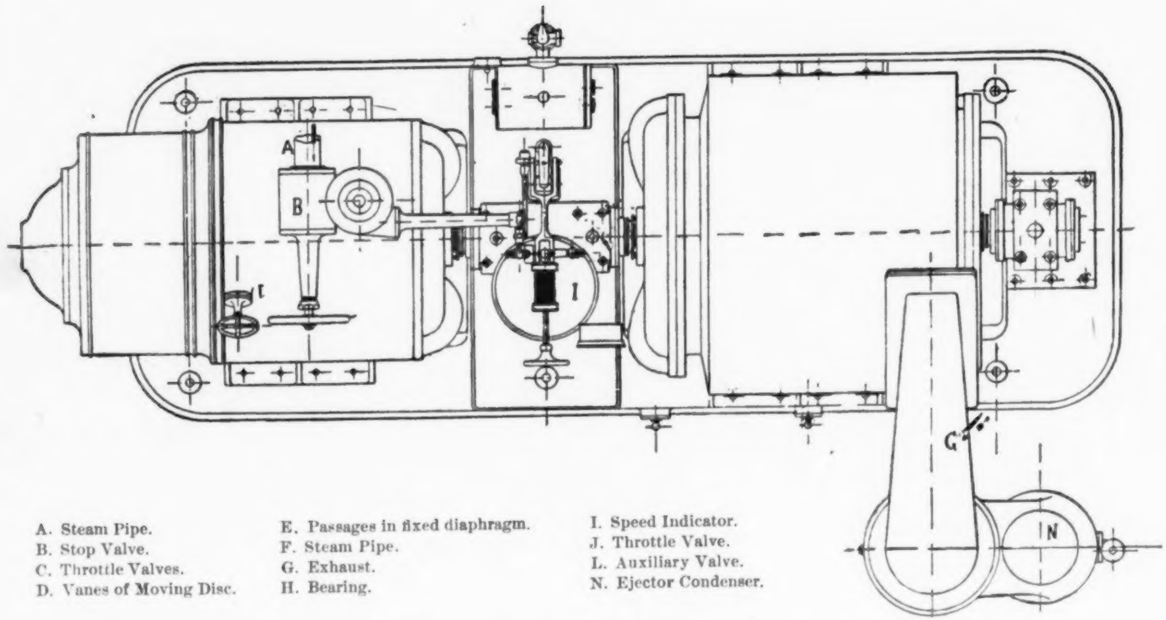


FIG. 393.—500 ELECTRIC H.P. TURBO-DYNAMO. 2,400 REVS. (Peñarroya.)

Longitudinal Section.



A. Steam Pipe.
B. Stop Valve.
C. Throttle Valves.
D. Vanes of Moving Disc.
E. Passages in fixed diaphragm.
F. Steam Pipe.
G. Exhaust.
H. Bearing.
I. Speed Indicator.
J. Throttle Valve.
L. Auxiliary Valve.
N. Ejector Condenser.

FIG. 394.—PLAN. 500 ELECTRIC H. P. TURBO-DYNAMO. 2,400 REVS. (Peñarroya.)

consumption of steam at no load with the dynamos excited usually reaches from 20 to 25 per cent. of the consumption of steam at full load. It therefore follows that turbine sets are more advantageous than reciprocating engine sets when running at light loads, even supposing it be admitted that there is equality at heavy loads. Another valuable property of steam turbines is that they may be considerably overloaded without difficulty. Thus, in the case of the machines at Peñarroya, which were designed for 500 horse-power, they have been able to develop 650 electrical horse-power, and the output could have been raised even still higher had it not been

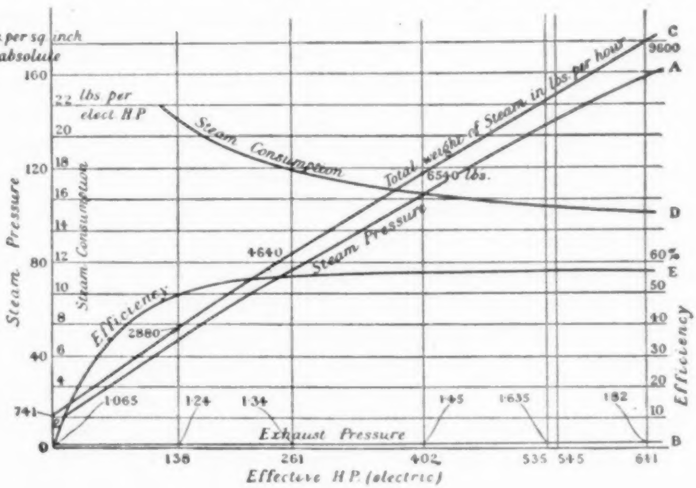


FIG. 395.

for the overheating of the commutators of the dynamos. In our machines we are enabled to take an overload by supplying steam to a distributor more or less distant from the inlet in the series of successive distributors. The stop valves which regulate this supplementary admission can be moved by hand or controlled mechanically by the governor. The second group was submitted to a competent committee (consisting of Professors Stodola, Wyssling and Farny of the Zürich Polytechnicum) and this commission found the figures for consumption of steam a little higher (from 2 to 4 per cent.), the ampère meter brought from Zürich having given somewhat higher readings than the instruments standardized by the central laboratory of the Ecole supérieure d'Electricité in Paris.

23. These sets of turbo-dynamos for direct current have proved

somewhat unsatisfactory in practice, owing to the sparking of the brushes occasioned by the vibration of the armatures. The commutators of direct-current dynamos are, as is well known, the delicate parts of these machines, particularly when they have to run at such a great angular velocity as is necessary if direct coupled to steam turbines. In order to obtain sparkless running special arrangements must be made for absorbing the vibration, but it is very difficult to maintain in good equilibrium such heterogeneous bodies as the armatures of dynamos consisting of bars of copper which become slightly displaced under the action of centrifugal force owing to the compression of insulating material. Special arrangements for static equilibrium, and also for dynamic equilibrium have been designed, but the construction of the armatures is still always somewhat delicate, and it is only after several improvements that it has been possible to make them work well. At Peñarroya the vibrations were caused principally by want of level between the bearings, and this was brought about by settlement of the masonry supporting the machine. In order that bodies in rapid rotation upon very long shafts supported by several bearings may turn easily without vibration, it is indispensable that the principal axis of inertia of each part should coincide with the axis of rotation, and the bearings must be in correct alignment. In spite of these difficulties it has been possible to build good direct-current dynamos for direct coupling to turbines, and when the voltage is high the problem is simpler. Owing to the high peripheral velocity of the commutator it is usually necessary to employ metallic brushes, but in certain cases, notably with turbines supplied by steam at low pressure when the angular velocity is lower it has been possible to use carbon brushes. The machine at Bruay, which will be referred to later on, is an example. Among other sets for direct current built by Messrs. Sautter-Harlé may be mentioned one of 500 horse-power for Huta-Bankowa (Russian Poland), which is interesting from the fact that the two portions of the turbine and the two dynamos were built on two parallel shafts. This arrangement offers the advantage that one of the halves of the group can be used if the other is accidentally damaged.

24. *Turbines with Alternators.*—The construction of generators for two-phase and three-phase current for direct driving by turbines offers much less difficulty than the design of direct-current dynamos for the same purpose. The absence of the commutator

renders possible higher angular velocities for alternators than for direct-current dynamos. Three types of alternators have been tried successively. The machine with the solid rotor, that with the rotating field magnet and that with the rotating armature. The first type would be, from a mechanical point of view, the ideal generator for coupling to a steam turbine. Unfortunately from its design it is impossible to use speeds higher than 1,500 revolutions per minute for a frequency of 50 periods per second. On the other hand, this type of machine permits much greater magnetic leakage than the others, and this necessitates disproportionately large dimensions and a comparatively low efficiency. These circumstances are unfortunate, for the alternator with the solid rotor would permit of making the movable part of very solid construction which is easy to balance once for all and thus avoid any chance of mishap. It is on these grounds that types of alternators with revolving field-magnates or revolving armature are in actual use, and Messrs. Sautter-Harlé have already built a great number of them. The view in Fig. 396 represents a turbo-alternator with revolving field-magnets to develop 400 electric horse-power at 5,500 volts, and this machine is now working in the generating station of the Loire Electricity Co. The moving part of this machine makes 3,000 revolutions per minute. The results of tests have been as follows:

Pressure of steam	170 pounds absolute.
Back pressure of the exhaust	2.85 pounds.
Output of the terminals	388 electric horse-power.
Consumption of steam per electrical horse-power, including excitation	19.2 pounds.
Combined efficiency	48.7 per cent.

25. The efficiency obtained could have been much improved by increasing to a slight extent the dimensions of the turbine which had only twelve moving wheels, but even now it is comparable with turbine sets of the same power of other types. Three similar sets working with superheated steam are now being constructed for the factories of Pavin de Lafarge at Teil (Ardèche). The results guaranteed by the makers are as follows:

Pressure of steam on admission to the turbine	156 pounds.
Vacuum in the condenser	26 inches.
Temperature of the steam	270 degrees C.
Consumption of steam per electrical horse-power at the terminals, including excitation	15 pounds.

All these different machines are fitted with ejector condensers.

26. A machine of 2,000 kilowatts at 1,500 revolutions per minute would have a turbine efficiency of 68 per cent., which would enable the following guarantee to be given, assuming the use of superheated steam and the condenser giving a very good vacuum, which it is easy to obtain with steam turbines in which there is no entrance for the air:—

Pressure of steam on admission to the turbine	200 pounds.
Vacuum in the condenser	29 inches.
Temperature of the steam	660F.
Consumption of steam per horse-power hour upon the shaft of the turbine	8.5 pounds.
Consumption of steam per kw. hour at the terminals	12.2 pounds.

This is a far better result, we believe, than can be given by reciprocating engines, and this can be obtained with steam turbines under the conditions specified.

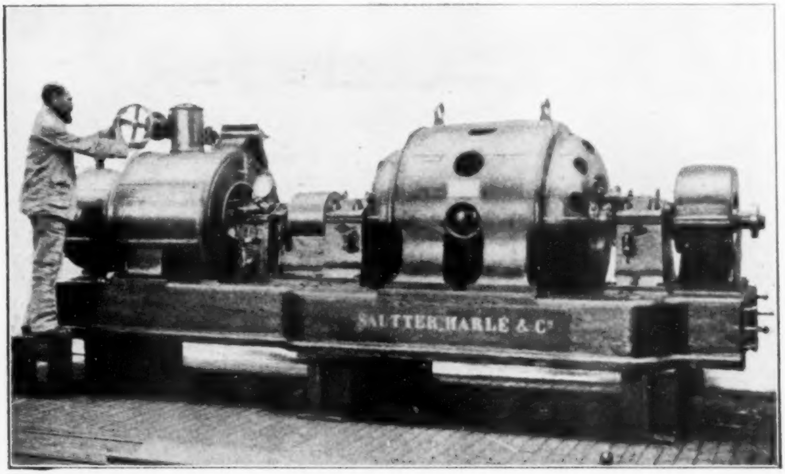


FIG. 396.—400 ELECTRIC H.P. TURBO-ALTERNATOR, 3,000 REVS., WITH REVOLVING FIELD MAGNET.

27. *Turbines for Vessels.*—The author read a paper on the 25th of March last in London before the meeting of the Institution of Naval Architects upon the application of steam turbines to the propulsion of vessels. He then described the difficulties which occur in the application of turbines to the propulsion of vessels which are as follows:—

(1) The difficulty of adapting screw propellers to the high speeds of rotation of the turbines.

(2) The poor efficiency of turbines at low velocity.

(3) The inconvenience of the combination in stopping and approaching quays.

28. The author believes that the best solution will be reached in the employment of a reciprocating engine of small power and of steam turbines coupled to independent shafts so that the reciprocating engine would always be working at any speed of the vessel. Then each type of machine would be perfectly adapted to the part which it has to play and excellent results in consumption of fuel would be obtained at all speeds. The author gives below a short account of the actual results obtained by turbines in the propulsion of vessels.

29. *Torpedo Boat* 243.—The equipment of the French torpedo boat No. 243 with steam turbines was decided upon several years ago, in 1898. The French Admiralty wished at that time to make experiments upon turbines and upon the working of multiple propellers. In order to reduce the costs of this experiment as far as possible it was thought desirable to make use of the hull of an ordinary torpedo boat, but great difficulty was found in making room for the turbines, and moreover the propeller shafts were placed very high at the turbine end and were necessarily considerably inclined; in this case to 11 per cent. from the horizontal. The back supports had to be considerably increased, and therefore the total resistance of the boat was much greater. All these conditions are extremely unfavorable and the screws fixed upon the inclined shafts gave a very bad efficiency. The two turbines, each of a nominal output of 900 horse-power, are completely independent of each other. They are provided on the exhaust side with a single moving wheel for movement in a reverse direction. As the experiment was solely made to see if the installation of turbines with propellers of small diameter was suitable for producing in a vessel a speed comparable to that given by a reciprocating engine, the backward movement of the turbine was willingly sacrificed. Very careful experiments have been made with this machinery, and it will be seen from the table given below that a speed of 21 knots has been attained. It is quite certain that if the shafts had not been set at such a great inclination a speed of 24 knots would easily have been exceeded, for the turbines gave an output somewhat higher than was expected.

30. The turbines worked in a most satisfactory manner, as is proved from the numerous reports of the trials made under the control of the Engineers of the French Navy.

52

31. *A Vessel Built by Messrs. Yarrow & Co.*—The vessel of Messrs. Yarrow & Co. is similar to torpedo boats of the first class usually constructed by this firm and similar, with the exception of the turbines and propellers, to the "Tarantula" upon which Parsons turbines are used. The boat is of 140 tons burden, and is provided with three propeller shafts driven simultaneously and separately by a turbine divided into two portions, and a reciprocating engine. This latter engine develops 250 brake horse-power and drives a central shaft which is completely independent of the turbines. The side shafts are driven by a turbine in two parts, arranged in series and rotating in opposite directions. The re-

TABLE II.

TORPEDO BOAT NO. 243. TRIALS OF JANUARY 22, 1903.

Six Propellers: Diam., 23.6 in.; Pitch, 19.7 in.

Number of Trial.	I.	II.	III.	IV.
Speed of vessel (mean of three runs) knots.	17.07	19.59	20.94	21.26
Rotation of turbines......revs. per min.	1,348	1,572	1,748	1,774
Effective pressure of steam on admission to turbines...........lbs. per sq. in.	68.26	100.98	129.42	132.26
Condenser vacuum.................ins.	28	28	27	27.5
Mean slip of propellers...............	0.217	0.230	0.260	0.260

ciprocating engine was used by us in order to obtain a velocity of 10 to 14 knots in the best conditions of consumption of coal, and so that movement astern might be easily effected. The above combination adopted by Messrs. Yarrow & Co. has given the expected results; but it is not the only possible design, and the author thinks that it would have been even more advantageous to arrange the driving mechanism so that the reciprocating engine instead of allowing its exhaust to escape directly to the condenser should pass it into the turbine at low pressure and perhaps even into the turbine at high pressure during a slow speed run. The turbine of the Yarrow boat can develop more than 2,000 horse-power. The high-pressure portion is represented in longitudinal section in Fig. 397. At the points where the shaft passes through the frame tightness is obtained by a system of special packing, the same as that employed upon land turbines. A special governor assures constant pressure upon the four packings so as to prevent all entrance of air, and owing to

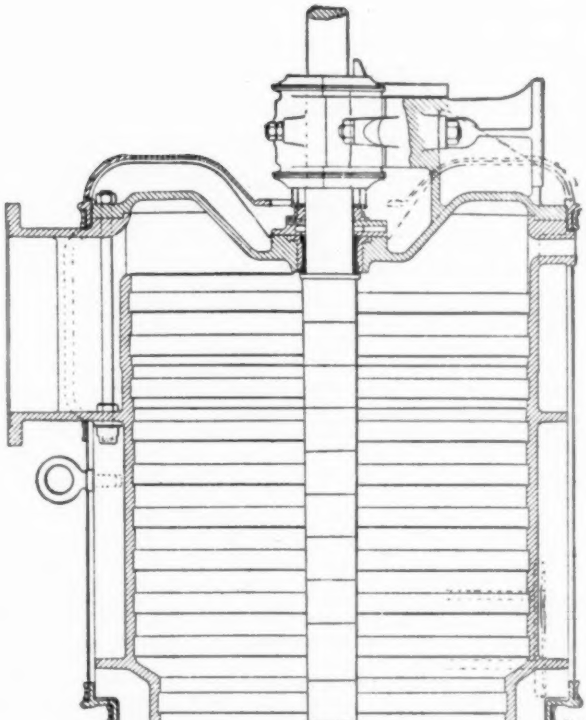


FIG. 397.—TURBINE FOR MESSRS. YARROW'S BOAT.

Longitudinal Section of H.P. portion.

this arrangement it is quite easy to obtain an excellent vacuum in the condenser. Several tests were made upon this vessel between the 13th October, 1903, when it first went out and the present date. The table of experiments made on the 19th January, 1904, after the propellers had been improved is given below. The speed of 25 knots which had been attained in the first trial of the boat has been increased to 26.39 knots by giving to the turbines a little more than the quantity of steam for which they were designed.

TABLE III.

MESSRS. YARROW & CO.'S TORPEDO BOAT. TRIALS OF JANUARY 19, 1904.

Number of Trial.	I.	II.	III.	IV.
Effective pressure of steam on admission to H.P. turbine........................lbs. per sq. in.	50	100	150	170
Condenser vacuum..... ....inches.	28	27.5	27	27
Speed of vessel (two runs).............knots.	15.58 20.00	19.25 23.53	23.22 26.67	25.714 27.067
Mean speed of vessel....................knots.	17.79	21.39	24.94	26.39
Rotation of reciprocating engine..revs. per min.	458	508	555	576
Rotation of H.P. turbine..........revs. per min.	836	1,052	1,207	1,258
Rotation of L.P. turbine..........revs. per min.	836	1,065	1,232	1,307
Slip of propellors: Reciprocating engine. per cent.	28.7	22.4	17	15.3
Slip of propellors: H.P. turbine........per cent.	13.6	17.4	16.4	14.8
Slip of propellors: L.P. turbine........per cent.	24.0	28.2	27.8	27.8

32. The propeller shafts were for this test each fitted with two propellers. The author's opinion on this subject is that the use of a single propeller upon the same shaft is certainly likely to give a better efficiency. The arrangement of propellers, one in front of the other is defective. The aft propeller works in the moving tail water from the former. It is certain that the speed of 26.4 knots can be still further improved and will become equal or even higher than the maximum speed given by reciprocating engines. The use of a single propeller for each shaft necessitates a peripheral diameter greater than the pitch in order to obtain sufficient propulsive surface. Under such conditions the efficiency can never be good unless the inclination of the shafts to the horiontal is very slight. In spite of this it is probable that the efficiency of these propellers with a relatively small pitch is distinctly less than that of ordinary propellers. Fortunately, this inferiority can be compensated by

the fact that the efficiency of well-designed turbines is a little higher than that of the reciprocating engines used on board ship.

33. *Turbine-Driven Pumps.*—The application of steam turbines for the purpose of direct driving of centrifugal pumps enables most remarkable results to be obtained and particularly renders possible height of lifts much greater than those which are at present obtained. It is well known that the pressure produced by the wheel of a centrifugal pump increases directly as the square of the peripheral speed, and therefore with the velocity produced by steam turbines it becomes possible with a single wheel to raise large volumes of water to heights of more than 900 feet in a single lift. The author, in the "Rapport au Congrès de Mécanique," in 1900 published the drawing of a turbo-pump of only one wheel enabling the attainment of a head of 700 feet and developing a useful power of more than 40 horse-power. The test of this pump and its driving turbine, made in 1900, was published in Ventilateurs et Pomps Centrifuges à Haute Pression," 1902. If it is desired to obtain either greater pressures or to obtain the same results with relatively lower speeds it is sufficient to couple steam turbines with pumps consisting of several similar wheels arranged in series so that each one increases by an equal quantity the pressure already given by the preceding wheels. The author has himself designed a particular type of multi-cellular centrifugal pumps. These machines are designed upon the general ideas set forth in a memoir which he published in 1892 in the "Bulletin de la Société de l'Industrie Minerale." The principal results of the general theories established at that time were further developed in the "Treatise on Turbo Machines," which appeared in 1897-1900, in which the author showed their application to hydraulic turbines, pumps, and centrifugal fans. From a practical point of view, by combining the advantage of high velocities of rotation with the advantage obtained by putting the wheels in series, it would be quite easy to deliver to heights greater than 1,500 feet. Some notes upon this subject were given in a communication made in 1902 to the "Société d'Encouragement pour l'Industrie Nationale." At that time a considerable number of turbine-driven pumps had been manufactured by Messrs. Sautter-Harlé. Multi-cellular turbine pumps form very light sets, which are compact and extremely simple. Their maintenance is very low for the wear is almost nil and the cost of lubricating oil extremely small. In the examples referred to below it will be seen that the

consumptions of steam per brake horse-power in water raised are comparable with those of good piston pumps.

34. *Turbine-Driven Pump at Falkenau (Bohemia).*—The first machine of this kind which was made was delivered to the mines of Falkenau in Bohemia. This apparatus consisted of a single bedplate upon which were mounted a steam turbine with multiple wheels enclosed in the same casing, and a pump with four wheels also mounted in a single case. The regulation of speed was obtained by a centrifugal governor arranged in the base of the principal bearing, Fig. 398. The lubrication of the end bearing of the shaft on the pump side was effected by means of a little hydraulic Servo motor. A compensating piston enabled the longitudinal thrust received by the moving part of a pump to be exactly counterbalanced. This thrust is, moreover, inconsiderable owing to the special arrangement adopted for the wheels of the pump. Condensation of the steam is effected by means of an ejector-condenser on the system designed by the author. The results obtained in the different tests of this set may be shortly set forth as follows:

TABLE IV.

Admission pressure of steam P. lbs. per sq. in.	82.5	91.9	94.6	97.5	104
Exhaust pressure of steam p (vacuum 24.8 ins. of mercury) .... lbs. per sq. in.	2.47	2.47	2.47	2.47	2.47
Revolutions per minute	3175	3200	3200	3200	3280
Total height raised .... feet.	721	721	695	669	688
Discharge of water .... gals. per min.	405	492	551	608	665
Useful work in water raised .... H.P.	89	107.8	116.7	123.6	139.4
Theoretical consumption of steam per H.P. hour .... lbs.	11	10.7	10.62	10.5	10
Actual consumption of steam per useful H.P. of water raised .... lbs.	34.61	32.15	30.3	29.5	27.8
Total combined efficiency of the turbine pump	0.315	0.335	0.350	0.355	0.360

It appears from these figures that it is possible with such a turbine-driven pump to obtain combined efficiencies of 36.5 per cent., which corresponds to an efficiency for the pump of 67 per cent., and an efficiency for the turbine of 55 per cent.

35. *Turbo-Driven Pumps at Bruay.*—A more powerful turbo pump has been made and delivered recently to the Mining Co., of Bruay, in the Pas-de-Calais. This set can raise 950 gallons a

minute to a height of 1,200 feet. The complete set mounted above a small tank into which discharges the ejector-condenser, which is used for condensing steam, and from which is taken the suction of the principal pump. At the end of the shaft is a small pump, which is directly coupled to the turbine, and furnished water to the ejector-condenser at a pressure equivalent to a head of 18 to 20 feet. The results obtained with this pump at full load were as follows:—

Discharge in gallons per minute	980 gallons.
Total lift	1,195 feet.
Revolutions per minute	2,200.
Absolute pressure of the steam on admission to the turbine	101 pounds.
Absolute back pressure at the exhaust	1.61 pounds.
Power in water raised	359 horse-power.
Consumption of steam per horse-power hour in water raised	22.7 pounds.
Total net efficiency	42.5 per cent.

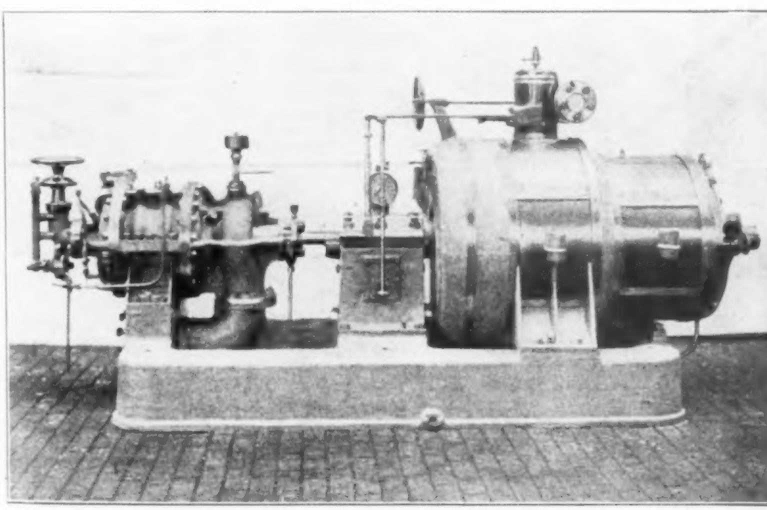


FIG. 398.—TURBINE DRIVEN PUMP AT FALKENAU (BOHEMIA).

36. It will be observed that the total efficiency of this set is very high, attaining to 42.5 per cent. It will be pointed out hereafter what a slightly improved efficiency denotes in consumption of steam when high pressures of steam can be used. One of the chief advantages of turbo pumps, besides their excellent efficiency, is

the extremely small floor space which they occupy when compared with piston pumps. Their dimensions are very small, and they require very little height, so that they can be placed in a room which has no greater dimensions than that of an ordinary working gallery in a pit. In Fig. 399 are shown the comparative dimensions of the chamber in the mine necessary to accommodate the turbo pump which we have just described, and of a chamber necessary to accommodate a piston pump of the same power. The cubic volume of the masonry work is approximately in the proportion of 1 to 10.

37. *Turbo Pumps for Boiler-Feed Purposes.*—This class of pump can be used very readily for feeding steam boilers, and several such sets have already been constructed. When there is a group of generators of the same size, a turbo pump can easily carry out the whole feed service with a consumption of steam much less than that of the small feed pumps usually employed. Whilst ordinary reciprocating pumps consume from 150 to 250 pounds of steam per brake horse-power in water moved, the turbo

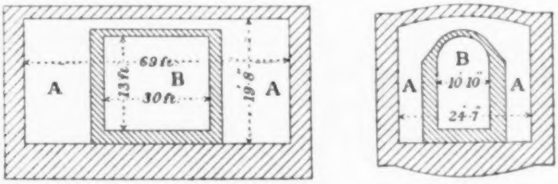


FIG. 399.—PUMP CHAMBERS AT BRUAY.

A, Reciprocating Steam-pump. B, Steam Turbine (Rateau).

pump for boiler-feed service will not usually require more than 40 to 60 pounds per brake horse-power. An automatic system of control has been designed by the author which enables the apparatus to work automatically in a continuous manner even when the demand falls to zero. The work of the stoker, so far as the feed service is concerned, consists simply in opening the feed valve to the necessary amount.

38. *Turbo Pumps of Large Output for Raising Water in Towns.*—Turbo pumps, both from the point of view of efficiency and also of ease in working, are particularly suitable for raising water into reservoirs for the service of towns. The following figures indicate results which can be obtained with this class of machine in similar conditions to those that exist in large towns.

Delivery in gallons per minute	5,200.
Height to which the water is raised	460 feet.
Actual horse-power in water raised	730 horse-power.
Combined efficiency including condensation	46 per cent.
Pressure of steam in pounds per square inch	210.
Vacuum at the exhaust	28 inches.
Super-heat in the steam	100 degrees.
Consumption of steam per horse-power hour in water raised	15 pounds.

It will be observed that the system described is quite as efficient as the best piston pumps now employed.

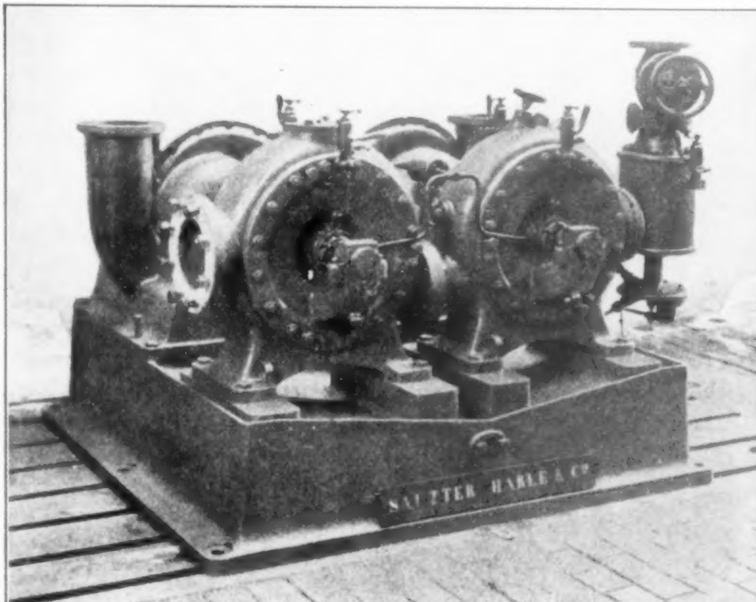


FIG. 400.—TURBO FAN FOR BLAST FURNACE AT COMMENTRY.

39. *Turbo Fans.*—The great angular velocity of steam turbines makes them particularly suitable for the direct driving of fans for high pressure used for blowing engines and even for air compressors. The author has already described in his work entitled "Fans and Centrifugal Pumps for High-Pressures" inserted in the "Bulletin de La Société de l'Industrie Minerale," of 1902, and has given particulars of actual results obtained from a turbo fan with a revolving wheel of 10 inches in

diameter. With an angular velocity of 20,200 revolutions per minute, this fan discharged 22.5 cubic feet of air per second under a pressure of 19 feet of water. The useful work contained in this compressed air was about 41 horse-power, and the combined efficiency of the group 30.7 per cent. The pressure of the steam on admission to the turbine was 145 pounds per square inch. Since that time the system described has been improved, and a certain number of these apparatus have been built for different purposes. Several are used in sugar refineries for the compression of air charged with carbonic acid extracted from the lime furnaces, and requiring to be sent into the decarbonizing vats. A special system of regulation enables the pressure of discharge to be kept constant whatever may be the volume of discharge, and on the contrary to maintain a constant discharge whatever may be the pressure. The set, which is shown in perspective in Fig. 400, is actually at work in the forges of Chatillon et Commentry, and is used for blowing a blast furnace. The set consists of two fans arranged in parallel and each driven by one turbine (the two turbines are placed in series) and the apparatus will discharge 80 cubic feet of air per second under a pressure of 10 inches of mercury. The energy in the compressed air rises as high as 100 horse-power. The moving wheels of the fans and of the turbines are only 11.2 inches in diameter, and the Table 5 gives an abstract of the results of trials to which the apparatus was submitted.

TABLE V.

TURBO FAN AT COMMENTRY. AUGUST 6–8, 1903.

Orifice.	Revs. per Minute.	Pressure of Steam Absolute. Admission.	Pressure of Steam Absolute. Exhaust.	Air-pressure at the outlet of Fan.	Total consumption of steam.	Air Discharge.	Horse-Power Theoretical.	Horse Power Useful.	Total Efficiency.
sq. ins.		lbs. per sq. in.	lbs. per sq. in.	Feet of water.	lbs. per hour.	Cubic feet per sec.			
19.1	15,200	113.8	4.53	10.60	3037	91.4	260.4	99	0.380
	15,700	128.0	4.80	11.61	3403	93.5	299.5	112.6	0.377
22.3	14,600	113.8	4.92	9.0	3037	99	256.9	93.2	0.363
	15,200	128.9	5.32	9.90	3403	103	289.6	106.52	0.368
	15,800	142.2	5.68	11.0	3760	107.5	327.2	121.91	0.372
27.8	14,300	113.8	4.44	7.55	3037	114	261.8	91.08	0.348
	14,900	128.0	4.80	8.25	3403	117.5	299.5	102.46	0.342

40. It will be seen from this table that the combined efficiency of the apparatus reaches the remarkable figure of 38 per cent. Similar apparatus of larger dimensions can be built, and these

might replace advantageously the large blowing engines used for blast furnaces. One important characteristic of these sets is their remarkably small dimensions, and they can be put down at an expense which is very small in comparison to large piston blowing engines. By coupling several wheels of centrifugal fans, as we couple the wheels of multi-cellular pumps, much higher pressures of air can be obtained. In a few months we shall finish the construction of a centrifugal turbine compressor of 350 horse-power capable of delivering compressed air at a pressure of 90 pounds per square inch. This apparatus will be driven by a low-pressure turbine using the exhaust steam from a winding engine used in a pit. The turbine has as a stand-by a high-pressure turbine so as to ensure economical working during the time when the winding engine is out of use. This special branch of the application of steam turbines to air-compression has, in our opinion, a great future, not only for work in mines, but also in metallurgy, where blowing engines of all kinds for blast furnaces and for the tuyeres of steel works may be replaced by the centrifugal machines of a much simpler and smaller character, costing far less and requiring less repairs than the present piston engines.

41. *Turbines for Low Pressure with Steam Accumulators.**— One of the most interesting and promising applications of steam turbines is the employment of steam at low pressure, and particularly exhaust steam coming from engines working intermittently. This last class of engines, which are, of course, usually employed in installations for the working of minerals or for rolling mills and pile drivers, has not until recently been able to benefit by the same advantages as is the case with engines working continuously, and this is owing to the special difficulties in the application of condensing plant, multiple expansion, and superheating to such engines. The majority of such engines discharge the exhaust freely into the air, and the quantity of steam which is thus lost is considerable. The author has given special attention to the interesting problem of the employment of such waste steam, and he has obtained satisfactory results by means of his regenerative accumulator of steam, combined with turbines at low pressure, which are themselves coupled directly to dynamos, centrifugal pumps, or fans. The regenerative accumulator is intended to regulate the intermittent flow of steam before it passes to the turbine, and it consists essentially of a vessel containing solid and

* Proceedings, 1901, page 945.

liquid materials which play the part of a fly-wheel for heat. The steam collects and is condensed as it arrives in large quantities in the apparatus, and is again vaporized during the time when the exhaust of the principal engine diminishes or ceases. The necessary variations for condensation and regeneration of the steam correspond to fluctuations in pressure in the accumulator, this pressure rising when the apparatus is being filled and descending when it is discharging into the turbine. Water which has a very high calorific capacity has been used as a heat fly-wheel, but in order to rapidly communicate to a liquid mass a considerable quantity of heat corresponding to the latent heat of steam to be condensed it becomes necessary, owing to the poor conductivity of water, either to arrange it in thin layers or to cause a rapid circulation in order to increase the surface of contact between the steam and the water itself. The first solution of the problem gave rise to the accumulator with flat cast-iron trays in which water is contained in shallow vessels arranged one above the other, Fig. 401. The second solution of the problem gave rise to the accumulator with water only in which a rapid circulation was produced by the injection of steam into the body of the liquid itself, Fig. 402. The low pressure turbine, fed by the regular flow which comes from the accumulator, and working, for example, between an admission pressure of 15 pounds per square inch and a vacuum at the condenser of 27 inches of mercury (back pressure of 1.6 pounds) can furnish an electric horse-power for about 31 pounds of steam per hour. In a moderate sized pit, consuming 13,000 pounds of steam per hour, it is, therefore, possible to economize under these conditions about 350 electric horse-power. In steel works, where reversible steam rolls are employed consuming about 45,000 pounds of steam per hour, it will be easy to develop, by means of accumulators and turbines, an extra output of over 1,100 electric horse-power. It is desirable to mention that the prime movers are in no way injuriously affected in their working by the presence of the accumulator and turbines. This method was applied for the first time at Bruay, and the installation has worked most satisfactorily since August, 1902, when it was first put into use. The exhaust steam from a winding engine is first of all treated by an accumulator with cast-iron trays, and then passes to a low-pressure turbine of 300 horse-power, which itself drives two dynamos keyed upon the same shaft; this group is shown in Fig. 403. A particular feature of this type of

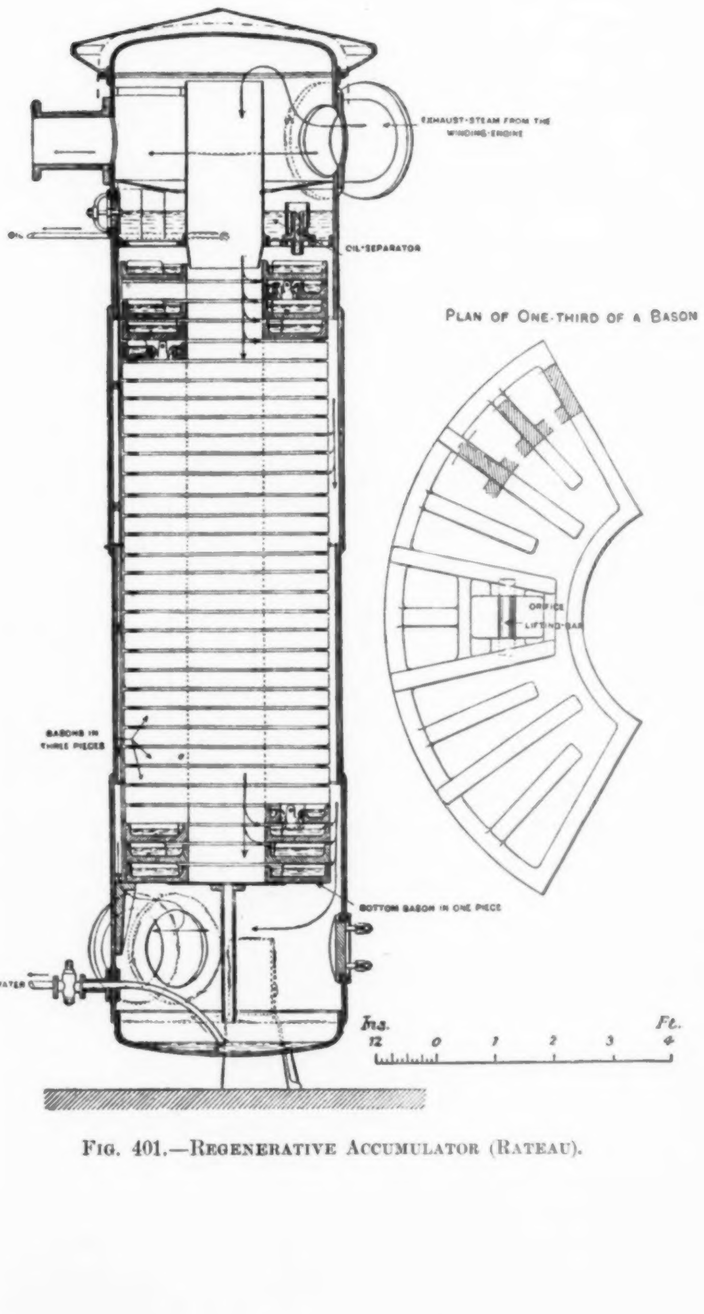


FIG. 401.—REGENERATIVE ACCUMULATOR (RATEAU).

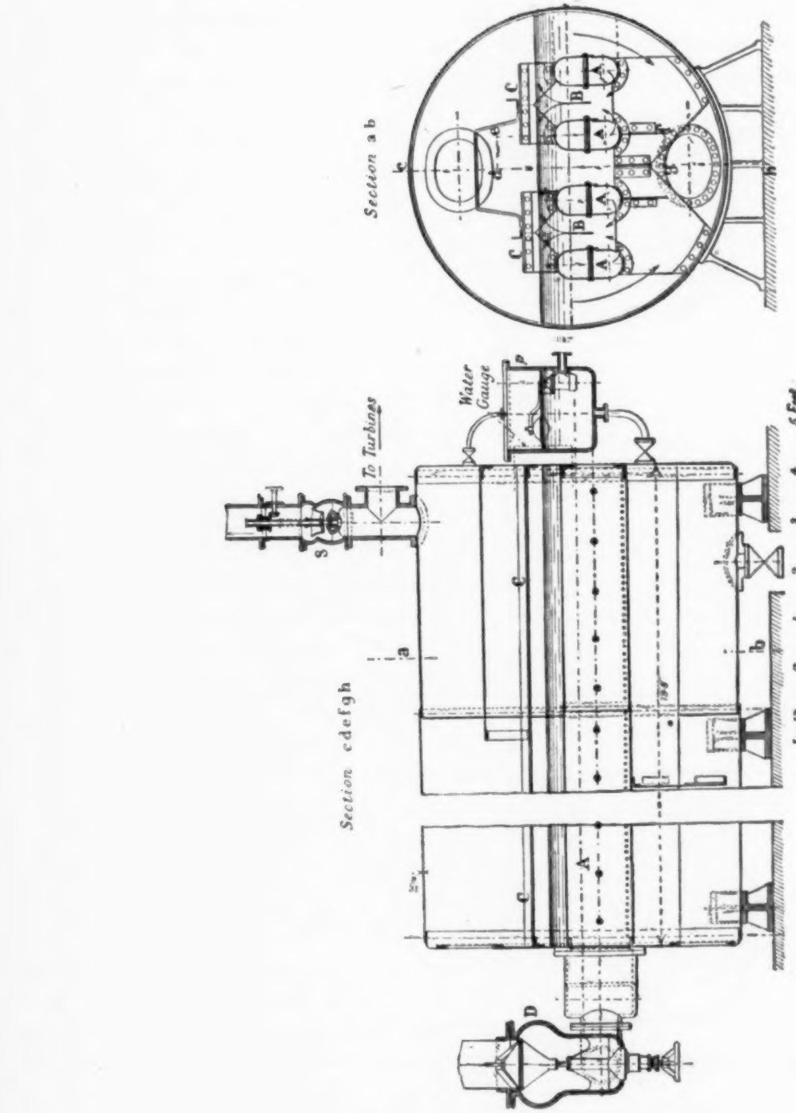


FIG. 402.—WATER HEAT-ACCUMULATOR (RATEAU).

installation is the "Automatic Expander," which comes into use when the winding engine is not giving sufficient steam, and also when this engine is not working. This apparatus then permits the admission to the turbine of live steam which is passed from the boilers through reducing valves; this expander can be adjusted by means of a spring, and comes into use as soon as the pressure falls below a pre-determined point.

42. Table 6 gives the results of tests made upon this set in April, 1902, and January, 1903. Numerous similar tests have been made at different times since the apparatus was first started, and Fig. 404 enables a comparison to be made between the efficiencies which were calculated from the results obtained in April, 1902, January, 1903, and November, 1903.

TABLE VI.

EXPERIMENTS WITH THE LOW-PRESSURE TURBINE AND DYNAMO OF 300 HORSE-POWER FOR THE BRUAY COLLIERIES.

Revolutions per Minute. n	Kilowatts at the Terminals.	Absolute Pressures: Steam to Turbine. P	Absolute Pressures: Condenser. p	Temperature of Steam. $t°$		Flow of Steam Measured. I	Consumption of Steam per Horse-power-hour: Measured. K	Consumption of Steam per Horse-power-hour: Theoretical. K	Efficiency. ρ*
		Lbs. per sq. in.	Lbs. per sq. in.	Cent.	Fahr.	Cubic feet per hour.	Lbs.	Lbs.	
April 3, 1892.									
1,337	169.5	12.01	2.13	132	269.5		45.2	21.9	0.488
1,560	176.5	"	2.22	"	"		42.8	22.2	0.515
1,610	186.0	"	"	"	"	165	40.5	"	0.547
1,690	190.5	"	"	"	"		39.7	"	0.559
1,840	198.0	"	"	133	271.4		38.1	"	0.581
1,510	225.0	14.37	2.49	135	275.0		41.0	21.2	0.516
1,605	232.5	"	2.52	"	"		39.8	21.3	0.534
1,700	240.5	"	2.57	"	"	201	38.5	21.5	0.559
1,800	247.0	"	2.62	"	"		37.4	21.7	0.580
April 5, 1902. Experiments made by Messrs. Sauvage and Picou.									
1,589	70.3	5.42	1.25	111.0	231.8	79	51.1	25.1	0.492
1,600	140.9	9.37	1.82	135.0	275.0	130	45.1	22.3	0.530
1,591	202.0	12.83	2.32	137.0	278.6	177	39.7	22.1	0.531
1,598	232.5	14.70	2.79	147.0	296.6	203	39.5	21.9	0.555

* The efficiency is the relation between the electric power measured at the terminals of the dynamo, and the theoretical energy contained in the current of steam utilized for such conditions of pressure: this yield is therefore stated after the deduction of all losses in the turbine and in the dynamo.

43. It will be noted from this table that although the tests of November 1903 were made with an output much lower than the normal output of the turbine and for a part of the curve in which the efficiency varies rapidly, the mean point which these tests graphically represent is very distinctly higher than the curve of 1902. The combined efficiency of the turbo-electric group has, therefore, been maintained at an excellent value since the set was first put into use. When, as at Bruay, the low-pressure turbine

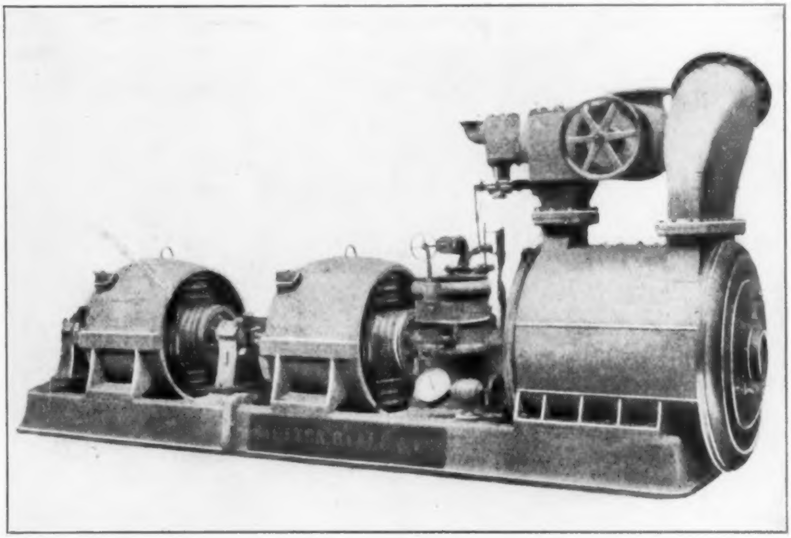


FIG. 403.—LOW PRESSURE TURBINE DRIVING TWO DYNAMOS (BRUAY COLLERIES.)

is fed with expanded live steam during the time when the prime mover is at rest, the consumption of steam is obviously higher than that of an engine working at the full pressure of the boilers. If the demand for live steam is only made during a relatively short period, the slightly higher consumption during these periods will have scarcely any influence upon the total economy. If this is not the case, and, on the contrary, such periods are of considerable length as compared to that during which exhaust steam is used, as, for example, when the prime mover is only working during the day, and it is desired to have a regular service day and night from the secondary set, then it would be best to add to the low-pressure turbine a high-pressure turbine which, during the time that the prime mover is not working, would receive live

steam at boiler pressure. The steam passing out of this high-pressure turbine would pass to the low-pressure turbine, which would use equally well either the steam coming from the accumulator or the exhaust from the first portion of the turbine. The admission of live steam at high pressure is, moreover, automatically obtained by a suitable system of obturators controlled by a speed regulator on the one hand and by the pressure in the accumulator on the other hand. Such a group works under all conditions with the maximum possible economy with an ordinary vacuum, and working at full load it would consume only 18 pounds of steam per electric horse-power hour if supplied with steam at 110

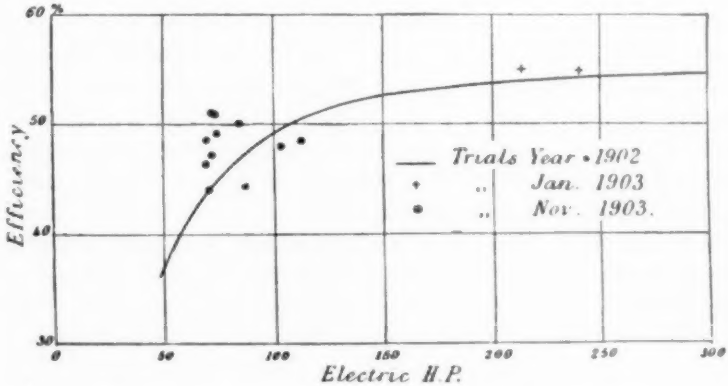


FIG. 404.—COMPARISON OF EFFICIENCIES (1902 AND 1903.)

pounds pressure, and it would consume 26 to 36 pounds of steam when it was fed by exhaust steam at atmospheric pressure. This system of regulation by means of steam accumulators and turbines, which is particularly applicable to mines and steel works, enables very considerable economy to be effected. Installations of this nature are now in course of erection at the following places:—

350 Horse-power at the Béthune Mine (turbine-driven air-compressor).
600 Horse-power at the Mines of Reunion in Spain (electric generating sets).
2,000 Horse-power at the Donetz Steel Works (electric generating plant).
700 Horse-power at the Acieries de Poensgen à Dusseldorf (electric generating plant, etc.)

As the author showed in a note published in January, 1904, in the "Revue Universelle des Mines," this system offers great

advantages when there already exists a central condensing plant, and the Poensgen installation is being carried out in that manner.

44. *Conclusion.*—There is no need to insist upon the well-known advantage in turbines resulting from their absolutely steady rotation, but, on the contrary, the enormous speed at which they must work in order to suit the velocity of flow of the steam is, in many cases, a serious inconvenience, particularly in the application of the turbine to the propulsion of vessels. It was a similar disadvantage for a considerable time in the driving of dynamos, but now it is possible to construct dynamos for very high speeds, and these can be coupled to steam turbines forming light and inexpensive sets which take up little room, although they are very powerful. Such sets are very simple and are very easy of maintenance. Besides the constant value of the turning effort, the chief advantages which can be claimed for steam turbines may be thus summarized: very low consumption of steam, particularly at small loads. From this point of view they are notably superior to piston engines of even the best-known types. Further advantages are the small floor space occupied and the absence of expensive foundation work; the cost of oil is very low, particularly for our turbines, which have oil reservoirs and ring lubrication to all the bearings. The oil in the reservoirs need be changed only once in two months, and it is not thrown out by the rotating parts, so that the machinery always remains clean. With steam turbines momentary or even permanent overloads are very easy to deal with. It is even quite possible to have overloads of more than 50 per cent. higher than the normal load. The regulation of speed in steam turbines is carried out in a more perfect manner than is possible for piston engines, and this is due, on the one hand, to the absolute constancy of the turning effort, and, on the other part, to the great kinetic energy accumulated in the revolving parts. This last characteristic makes the machines very insensible to variations in load. The full load may be taken off and put on again without causing a variation in speed of more than 2 to 3 per cent. When coupled to dynamos, fans or centrifugal pumps, steam turbines render it possible to obtain surprising results owing to their capability of giving great power at very high speeds. The author has constructed turbine-driven fans, giving a pressure of 7.5 pounds per square inch, and turbine-driven pumps raising water to several hundreds of yards. Such sets can no be built to develop several thousand horse-power, compressing air to more

than 90 pounds per square inch, or raising water to more than 2,000 feet high. Turbines are able to use to perfect advantage steam at low pressure, for they have an efficiency rising in value as the pressure of the steam becomes lower. In combination with steam accumulators of the type designed by the author of this Paper, turbines permit of the employment of exhaust steam in large quantities produced by engines working intermittently, such as winding engines and engines used in metallurgical works. In the preceding Paper, the author believes that he has given typical examples of the chief applications in which steam turbines are available. It will be seen that their field of application is very wide, and it is certain that a great future is in store for them.

DISCUSSION.

Mr. Francis Hodgkinson.—In Fig. 395 and Table I. Professor Rateau has given us a few details of some tests. He expresses the efficiency of his turbine by dividing the steam consumption of an ideal engine by the consumption actually observed.

This efficiency is undoubtedly the basis upon which engine performances may best be compared, but I rather take issue with Professor Rateau in the manner he has expressed this efficiency at fractional loads, insomuch that he appears to charge his machine with the heat value of the steam at the internal pressure in the cylinder, after it has been reduced by the governor valve instead of the pressure in the steam pipe adjacent to the throttle. I may be in error in this assumption, since no readings of boiler pressure are given.

The steam on being wire-drawn through the governor valve from presumably, in this particular test, a pressure in the steam pipe of about 156 pounds to the figure he has given of 46.21 pounds at one quarter load, would do much work in the way of superheating, such that the efficiency figures quoted would be naturally unfairly high.

I would calculate the efficiency at one quarter load to be about 39.2 per cent. instead of 51 per cent., assuming a pressure of 155 pounds absolute at the throttle.

The engineering world I believe will look forward with the greatest possible interest to the results of Professor Rateau's experiments to find the value of the specific heat of steam at

high temperatures and at other pressures than atmospheric, at which pressure Regnault made his experiments.

This country, however, has not been idle in the matter, as Professor Carpenter has lately concluded a large series of experiments with steam of higher pressures, and I believe Mr. Emmet has some information on the subject.

Mr. E. Meden.—In paragraph 2 Professor Rateau states that it is practically impossible to construct a turbine wheel to run at a peripheral velocity above 1,200 feet. If Professor Rateau had been familiar with the De Laval turbine, he would have known that turbine wheels, running at a velocity of over 1,200 feet per second, have been in successful operation at least since 1897.

Professor Rateau further in the same paragraph states that efficiency of a single wheel is necessarily low, chiefly due to the necessity of employing diverging nozzles, giving rise to great loss of energy due to friction, etc. These losses of energy undoubtedly occur, but the comparatively efficient results obtained from this type of turbine does not warrant Professor Rateau's assumption.

In the last part of paragraph 2 Professor Rateau speaks of the necessity of employing gears of special construction which are subject to excessive wear and accidental breakage. I do not understand what is meant by "gears of special construction," as the De Laval type of gearing is very old, and with the method employed for the manufacture of these gears, the cost is not at all excessive.

As regards wear, many of these gears have been in operation for upwards of nine years without showing any appreciable wear.

Regarding accidental breakage, we have yet to learn of a piece of machinery which is not subject to this misfortune

*Prof. A. Rateau.**—My lack of familiarity with English has precluded my active participation in the discussion in a public way, and I have for this reason regretted that I have been unable, as would have been appropriate, to express my thanks for the sympathetic reception which has been accorded to my paper on the steam turbine. It is particularly agreeable to an investigator to bring his work before a meeting of engineers and scientists in America, who can appreciate better than all others the services which science is rendering each day to industry. To prevent my

---

* **Author's closure, under the Rules.**

communication from becoming too technical and dry, I have avoided any theoretical development of the question. Those who would desire to go further than this will find certain studies of mine presented hitherto, in the "Bulletin de la Société de l'Industrie Minerale à St. Etienne," or in the "Revue de Mécanique" (Paris).

I will reply first to the criticisms made by Mr. Meden. This will serve to bring out certain points only touched on in my communication.

Mr. Meden has stated that if I had been familiar with the De Laval form of turbine I would know that wheels turning at a velocity in excess of 1,200 feet per second, have been in operation since 1897. My reply is that I do not ignore this fact, particularly since this result could only be attained by the use of that form of moveable wheel which I have been producing in Paris since 1895. We have ourselves caused wheels to revolve at higher speeds than this, provided they were made from one piece of metal and that the vanes were made veryshort and milled out from the solid block of metal. In my opinion, although ruptures of these wheels have been little to be feared, I hold that such high velocities are not to be recommended in practice. I consider these speeds are dangerous when the vanes are attached to the rim, and are relatively long, as is the case in the De Laval design. I had never seen the De Laval Turbine when my communication was prepared. I only gave the result of my personal experience.

Since Mr. Meden pleads for the turbine of De Laval I may be, therefore, allowed to say that I know of a considerable number of breakages of this type of machine. I will only recall the accident relatively recent at this writing, where the explosion of such a turbine has caused the death of several persons at Seraing in Belgium. This form of accident is apparently caused by the rupture of one or of several vanes after the steam has worn them. There follows an eccentricity of the revolving disc, which by reason of the very high velocity of rotation brings about the rupture of the shaft. Sometimes the disc remains in the chamber which surrounds it and turns there for a time more or less prolonged. But it also happens here that this chamber is broken and a species of explosion follows.

I know even a case where the disc kept on revolving at some distance from the chamber, turning on one of the ends of the shaft like a Dutch top. It is without doubt to mitigate the intensity

of these tendencies that certain builders of turbines of the De Laval type have determined to make their chambers of cast steel rather than of simple cast iron. It is for these reasons that I regard my statement justified that it is not possible " in a practical way " to exceed to any notable extent the peripheral speed of 1,200 feet per second. Furthermore Mr. Meden says himself in his paper on the De Laval turbine presented at this same meeting, that the ruptures of vanes occur more frequently in large turbines than in small ones. This must surely result from the higher peripheral speed.

The writer is versed in the question of the resistance of a revolving body. There will be found under his name a study on this subject in 1890 in the " Bulletin de la Société de l'Industrie Minerale a St. Etienne." In this work he showed that discs of uniform thickness should begin to rupture at the center, and Mr. Meden has pointed out exactly certain experiments which show that this condition will establish itself. He has further shown that greater velocities at the circumference can be attained by utilizing the radial strength of the material, that is to say by using discs which are enlarged at their center. The illustration Fig. 389 in Mr. Meden's paper of the De Laval design subsequent to 1897, shows exactly the form of flexible wheel made under my specifications since 1895 by the shops of Sautter-Harlé at Paris, except that our blades are milled in the rim while the De Laval design shows blades attached by a tenon. In 1895 the form and construction of the De Laval wheel were quite different.

The wear of the blades in the current of steam which is insignificant in multiple turbines, is on the contrary sufficiently rapid in the turbine of a single wheel, or of two or three wheels only. This follows from the greater velocity of the current of steam. This wear appears to be due principally to the shock of small particles of water which the steam contains after its expansion in the nozzles. It is diminished by utilizing superheated steam.

This liquid friction is a cause of important lessening of the output, besides being the occasion which leads to the rupture of the vanes after a certain time, as stated above.

Mr. Meden criticises me for having designated the gearing of high speed turbines by the epitaph " special." I did not intend to use a term which could have given offence even to those who are sensitive. These high speed gearings have this in particular that the teeth beside being helical are of very fine pitch. They

demand, therefore, a very exact construction. They are cut by machines specially constructed for this purpose, I should suspect, and not by general or ordinary types. Furthermore, after cutting they are rectified by grinding them with emery powder. This would appear to me to constitute a special construction. It is not a matter of much consequence if the cost of this construction is not excessive, and Mr. Meden says it is not. Some reference has also been made in a summary fashion to the question of losses in divergent nozzles. I have already given certain explanations on the subject in the number for September, 1903, of the "Revue de Mécanique." In his communication Mr. Melden describes the turbo-pumps of De Laval and presents certain results of experiment. May I profit by this occasion to recall that I have been, I think, the first to show that very high elevations might be secured with centrifugal turbo-pumps.

My first experiments go back to the year 1900, and I published them in the "Bulletin de la Société d'Encouragement" of 1902. Furthermore the graphic curves given by Mr. Meden show the economy as a function of the discharge of the pump are manifestly inspired by the more general curves, which I have pointed out under the name of "Characteristic Curves of Centrifugal Pumps."

Our first turbo-pump was little else than an experimental apparatus. Its high speed of rotation made it a machine of little significance to industrial uses according to our opinion, by reason of its delicacy. And we have produced a more satisfactory type by using turbines and multi-cellular pumps. The economy is thus much better and we avoid completely the use of gearing, which is the delicate part. My communication refers to turbo-pumps of this latter type. We obtain with them a combined economy of pump and turbine of more than 42 per cent., if judgment can be based on the figures which I have given for a pump of 350 effective horse-power.

The curves given by Mr. Meden in his Fig. 397 for a pump of 55 horse-power exhibit a maximum economy of 21 per cent. only. This is one-half of the economy of my design, although allowance should be made for the smaller size.

The better result is in fact a consumption of 41 pounds of steam per horse-power hour in water lifted, with a pressure of the steam of 180 pounds per square inch, and a vacuum of 25 inches. Under these conditions the theoretical consumption should be 8.72 pounds

of steam per horse-power hour, whence a total economy would result $\frac{8.72}{42}=0.213$.

For a similar capacity of 55 horse-power the turbines with multiple wheels should easily attain a combined economy of 36 per cent., instead of 21 per cent.

I find myself in complete accord with Mr. Hodgkinson in his statement that to estimate the net economy of the combined machine it is wise to start from the pressure of steam in the pipe, in advance of the throttling valve, rather than from the pressure after the throttle. That second pressure can only be used as I have done it if there is only in view the economy of the turbine itself independent of its method of regulation. It is easy to pass from one system to the other if the two pressures are known before and behind the throttle. I regret that I did not give in the tables the pressure in front of the valve. In the case examined by Mr. Hodgkinson this pressure was very nearly equal to that which he states.

No. 1039.*

LOCOMOTIVE TESTING PLANTS.

BY W. F. M. GOSS, LA FAYETTE, IND.

(Member of the Society.)

1. *Experiments of Alexander Borodin.*—In 1881 and 1882, Alexander Borodin, engineer-in-chief of the Russian South-western Railways, made an elaborate series of tests upon several small locomotives, blocked clear of the track, and so arranged that the power developed was absorbed by the shop machinery.† Three locomotives were tested, a simple engine having steam-jacketed cylinders, a simple unjacketed engine, and a compound jacketed engine.

In the process of testing, each locomotive in its turn was detached from its tender, placed outside and at right angles to the wall of a machine shop, in which position the locomotive, with its attached apparatus, was protected by a temporary roof. The driving-wheels were raised slightly above the rail, the coupled wheels disconnected, and the main driver on one side belted to a counter-shaft (Fig. 405). The maximum load which could in this manner be imposed upon the locomotive was about 90 horse-power, and because of this limitation, all tests were run under short cut-offs and low steam pressures. The cut-off was between 20 and 30 per cent. of the stroke and the steam pressure varied from about 60 pounds with simple engines to about 100 pounds for compounds. The diameters of the pulleys carrying the belting were so chosen that the revolutions of the locomotive driver corresponded with a speed of about eighteen miles per hour. In order to keep the power of the locomotive constant the minimum power required

---

* Presented at the Chicago meeting (May and June, 1904) of the American Society of Mechanical Engineers, and forming part of Volume XXV. of the *Transactions*.

† "Experiments on Steam-Jacketing and Compounding of Locomotives in Russia," by Alexander Borodin. Proceedings of the Institution of Mechanical Engineers, London, 1886.

to drive the shop was taken, and excess demands supplied by the regular shop engine. As the exhaust steam was condensed and could not be used to produce draught, the stack was lengthened to give the desired rate of combustion. The duration of each test was from one and a half to three and a half hours.

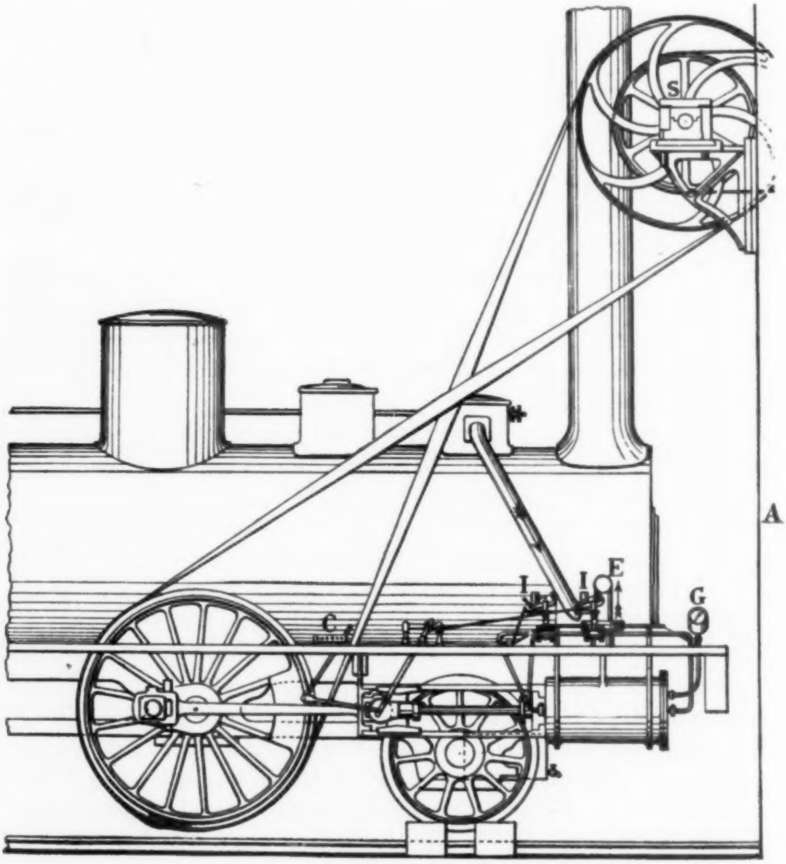


Fig. 405.

The locomotive under test was supplied with a steam gauge showing boiler pressure, a gauge for the steam jacket, a revolution counter, and one indicator on each cylinder. The weight of feed-water supplied was determined by use of a calibrated tank, and the overflow from the injector was caught and weighed. As a check upon the record of water consumption, and to determine

the amount of moisture in the steam, the exhaust steam was condensed, and its weight and heat determined. Six assistants were employed in taking observations.

A large number of tests were made under various conditions, and it is of interest to note that the results of this early work demonstrated the economy both of the steam-jacket and of the compound engine. The consumption of moist steam per horsepower per hour was found to be as follows:

Simple engine with steam jacket at 30 per cent. cut-off (pounds)	29.19
Simple engine with steam jacket at 20 per cent. cut-off (pounds)	26.98
Simple engine without steam jacket at 30 per cent. cut-off (pounds)	33.11
Simple engine without steam jacket at 20 per cent. cut-off (pounds)	32.16
Compound engine without steam jacket at 28 per cent. cut-off of high pressure cylinder (pounds)	24.92

Among the results of the tests were the conclusions that steam-jacketing affects:

1. A decrease in quantity of steam condensed during admission.
2. A decrease in the re-evaporation during expansion, and,
3. An increase of mean effective pressure.

The work of Mr. Borodin is now noteworthy because of the care with which the tests were planned and of the evident skill with which they were executed. It can hardly be said that the facilities he employed constituted a locomotive testing plant, and yet the considerations which guided him in his work were identical with those which a few years later led to the construction of a permanent plant. Mr. Borodin appreciated the necessity for operating a locomotive as a stationary engine for the purpose of testing its performance, and it is to this fundamental requirement that the modern locomotive testing plant responds.*

II. *The First Plant of Purdue University.*—The locomotive testing plant of Purdue University was the outgrowth of natural conditions. In the year 1890 the University was in the process of establishing an engineering laboratory to serve in the instruction of students and in the promotion of engineering research. Attempts to test stationary engines under the fluctuating con-

* It is but just to say that the later development of the testing plant in America, in no wise depended upon or grew out of the work which has been described. Two years after the installation of the first Purdue plant, M. Borodin personally inspected its operation, and it was on the occasion of this visit that the writer first learned of the work which had been accomplished ten years earlier by this distinguished Russian engineer.

ditions of service had proven unreliable, and experimental stationary engines so mounted as to permit their operation under constant conditions had already been given a place in several laboratories. It was easy for the Purdue authorities to foresee the great advantage to be derived from a study of locomotive performance, under conditions as favorable to the work of testing, as those which had been recognized as necessary in tests conducted upon stationary engines. The field was a promising one, because under conditions of service locomotives were tested with difficulty, and results obtained were even less satisfactory than those obtained from stationary engines under service conditions If the stationary engine was to be improved by the establishment of experimental engines, the locomotive could be vastly improved by the installation of experimental testing plant. By arguments such as these it was made clear that the process of building up the equipment of an extensive engineering laboratory might naturally involve a locomotive testing plant.

In May, 1891, an order was given the Schenectady Locomotive Works for a 17-in. by 24-in. eight-wheel engine, and in September of that year the locomotive, which had been named for its builders, "Schenectady," arrived at a switch a mile distant from the laboratory. There was no track which led to the laboratory. The surface of the ground to be traversed was slightly rolling, and a considerable portion of it was under cultivation.

In getting the engine over this ground, three sections of track were made, each of a rail's length, built in the form of skids and capped by 56-pound rails. The foundation of each consisted of two five by twelve yellow pine pieces, laid flat-wise, across which two by twelve pieces were spaced as ties, the rail spikes passing through the ties and into the foundation beneath. The working force employed in moving the locomotive included three pairs of horses with drivers, which with two or three men to handle blocking at the low places, furnished all the help necessary. One pair of horses by means of block and tackle was employed to give forward movement to the engine, a second to draw the skids one after another from rear to front, and the third to pull the heel of the advancing skid into line with the one previously placed. The men soon became so skilled in their respective parts that where the ground was smooth the engine could be kept in constant motion for considerable distances, one skid being drawn from rear to front and placed in position, while the engine was passing over the

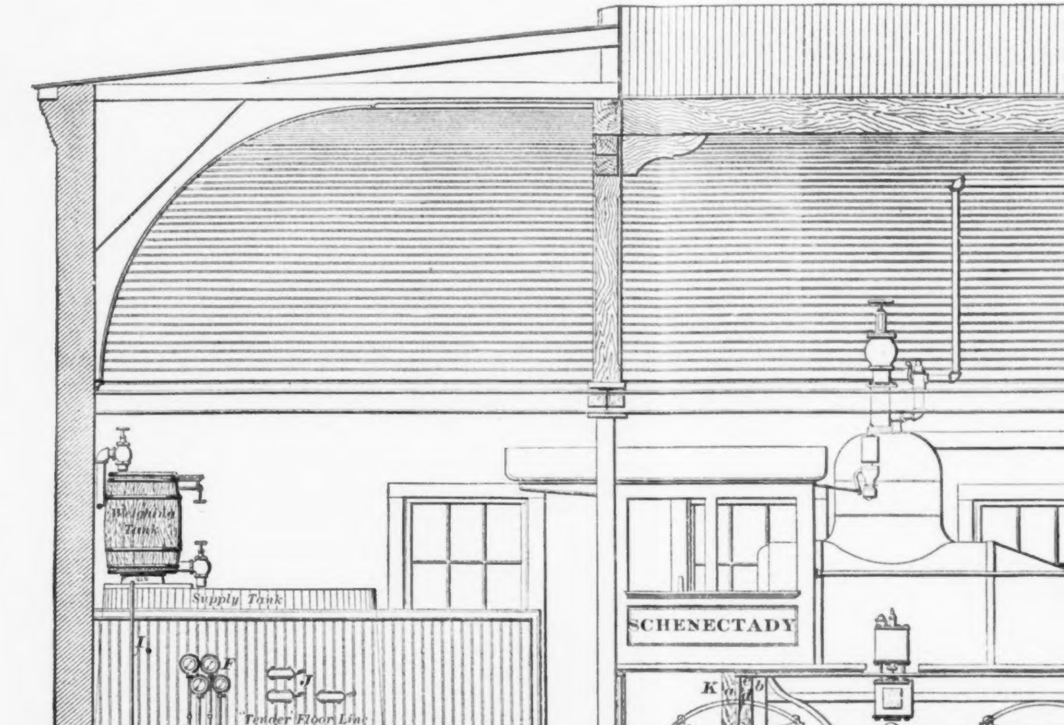
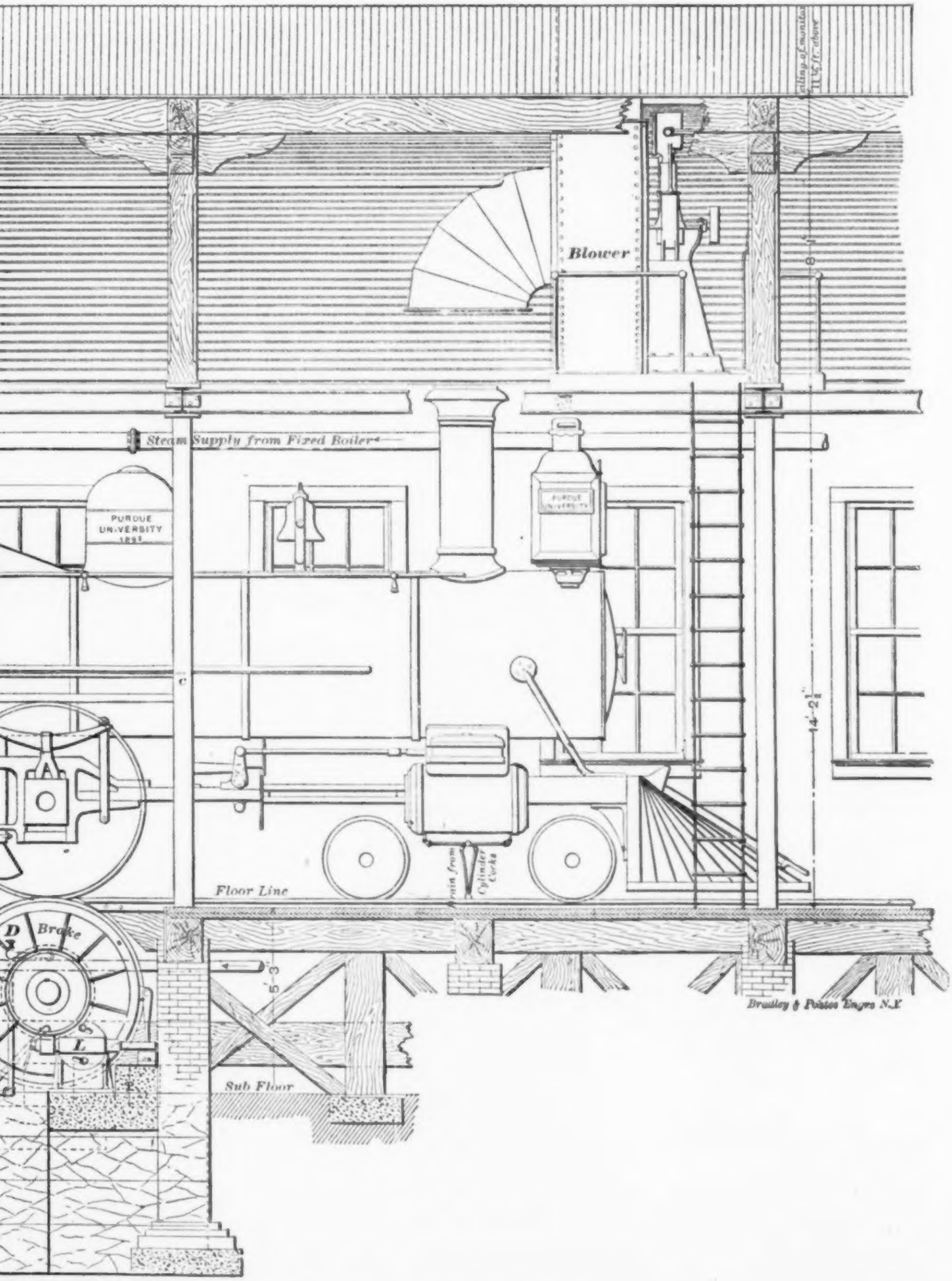


Fig. 406.—First Purdue Plant, Elevation of Loc

Reproduced from Vol. XIII, Tra



OF LOCOMOTIVE AND MOUNTING MECHANISM.

XIII, Transactions.

other two. The only difficulty experienced was in making the locomotive follow the skids if laid on a curve. This could not be accomplished. Wherever a change of direction was necessary it was made by laying cross-blocking under the skids, upon which one end of the skids bearing the engine was slipped bodily The course involved four turns, each somewhat less than a right angle, and the whole distance traversed by the engine was in the neighborhood of one and one-half miles. On the eighth working day after the start, the engine arrived at the laboratory without accident and without having once touched the ground.

Between the time when the locomotive was ordered and its delivery, the details of the mounting mechanism were designed and put in place, so that when the locomotive arrived the plant was practically ready for its reception. The completed plant, which is believed to be the first of its kind, is described in detail in a paper entitled, "An Experimental Locomotive," * which was read before the Society in 1892, from which Fig. 406 has been reproduced. Fig. 407 is from the photograph, showing the position of the locomotive in the laboratory.

The Purdue plan for mounting a locomotive for experimental purposes involved (1) supporting wheels carried by shafts running in fixed bearings, to receive the locomotive drivers and to turn with them; (2) brakes which should have sufficient capacity to absorb continuously the maximum power of the locomotive, and which should be mounted on the shafts of the supporting wheels; (3) a traction dynamometer of such form as would serve to indicate the horizontal moving force and at the same time allow but a slight horizontal motion of the engine on the supporting wheels. Assume an engine, thus mounted, to be running in forward motion, the supporting wheels, the faces of which constitute the track, revolving freely in rolling contact with the drivers. The locomotive as a whole being at rest, the track under it (the tops of the supporting wheels) is forced to move backward. If now the supporting wheels be retarded in their motion, as, for example, by the action of friction brakes, the engine must as a result, tend to move off them. If they be stopped, the drivers must stop or slip. Whether the resistance to be overcome in turning the supporting wheels is great or small the force to overcome it is transmitted from the driver to the supporting wheel,

* *Transactions* of the American Society of Mechanical Engineers, 1892, Volume XIII., page 427.

and will re-appear as a stress on the draw-bar, which alone holds the locomotive to its place upon the supporting wheels. The dynamometer constitutes the fixed point with which the draw-bar connects and serves to measure stresses transmitted. It is

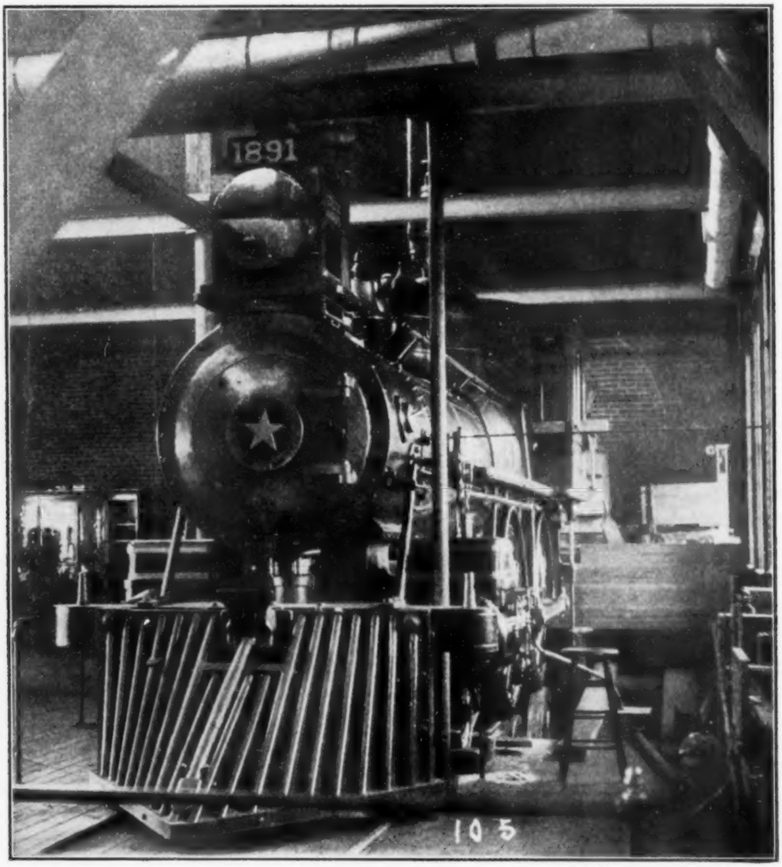


FIG. 407.—FIRST PURDUE PLANT.
General View.

evident from these considerations that the tractive power of such a locomotive may be increased or diminished by simply varying the resistance against which the supporting wheels turn, and that the readings of the traction dynamometer will always serve as a basis for calculating work done at the draw-bar.

Perhaps the most difficult problem in the working out of the

details, appeared in the design of the friction brakes, for not only were these required to be of large capacity, but the success of the whole plant depended upon their satisfactory action. They were constructed from drawings made at the University upon principles developed by Prof. George I. Alden, which had been described by him in a paper before the Society.* As they have once been described in the Transactions,† further reference to them at this time is unnecessary. The drivers of the locomotive were mounted upon supporting wheels with flat treads, having the inside edges rounded as in a rail. A traction dynamometer, made up of a system of levers and connected with the draw-bar, measured either the pull or the push exerted by the locomotive. A valve, actuated by one of the levers of the dynamometer, varied the supply of water to the brakes, thus automatically regulating the balance of the dynamometer.

During the school year of 1891 and 1892, following the installation of the plant, work upon it was directed more to the perfection of mechanical features than to the acquisition of scientific data. Nevertheless, during this year twenty efficiency tests were run, many of them at light power, and almost all with the throttle only partially open, all of which were later made the basis of a paper before the Society.‡

During the following year, in the course of a study of the counterbalance problem, the fact that a driver, will, under certain conditions, actually leave the track, through the action of its counterbalance, was demonstrated by passing wires under the moving wheel, some of these coming out with a portion of their length untouched by the wheel. The fact that this work involved speeds of from sixty to sixty-five miles per hour, was accepted as evidence of the practicability of the testing plant.§

Another series of experiments upon this plant was designed to show the effect of long indicator pipes, such as were then generally used in road tests of locomotives, the conclusion being that

---

* "An Automatic Absorption Dynamometer." *Transactions* of the American Society of Mechanical Engineers, Volume XI., page 959.

† *Transactions*, Volume XIII., page 427.

‡ "Tests of the Locomotive at the Laboratory of Purdue University." *Transactions* of the American Society of Mechanical Engineers, 1893, Volume XIV., page 826.

§ "An Experimental Study of the Effect of the Counterbalance in Locomotive Drive-Wheels upon the Pressures between Wheel and Rail." *Transactions* of American Society of Mechanical Engineers, 1894, Volume XIV., page 305.

the presence of such a pipe augmented the size of the indicator card, the distortion increasing with the increase of speed.*

Following this came a very complete series of efficiency tests, made under full throttle, and some of them at high speeds, the indicator showing from three hundred to six hundred horse-power. Much was expected from the data obtained from these tests, but it was destined to serve no useful purpose. On the 23d of January, 1894, the Engineering Laboratory, including the locomotive testing plant and all unpublished data, was burned.

If, in reviewing the results obtained in the two years or more, during which the plant was in operation, the facts developed appear all too few, it should be remembered that at first much time was spent in getting the plant into a condition of smooth running. Again, the locomotive was merely one piece of apparatus used in the instruction of students, and could receive attention only for its share of the time, and, consequently, was often idle; its operation was further limited by the cost of fuel and oil. But the few results which were obtained are not without significance.

III. *The Second Plant of Purdue University.*—The fire entailed a heavy burden of labor and expense, but with it there came also new opportunities. All details of the mounting mechanism were most carefully reviewed, and every fragment of experience was made to serve a useful purpose in the arrangement of a new plant. The apparatus was housed in a separate building, designed especially to receive it. A permanent track, 8,000 feet in length, was laid to connect the laboratory with the railways of the country. The damaged locomotive was extricated from the ruin, sent out over the track and thence to Indianapolis for repairs. Upon its return, a few weeks later, it was put under its own steam and backed in over the Purdue track directly to its place upon the supporting wheels of the new testing plant. The ease and rapidity with which this trip was made were in striking contrast with the laborious methods which attended its first trip across the same territory. Four months after the fire the new work had been completed and the reconstructed engine was in position.

The new plant (Figs. 408, 409, 410 and 411) embodies many important changes. While the first plant was designed as the mount-

* "The Effect, upon Diagrams, of Long-Pipe Connections for Steam Engine Indicators." *Transactions* of American Society of Mechanical Engineers, 1896, Volume XVII., page 398.

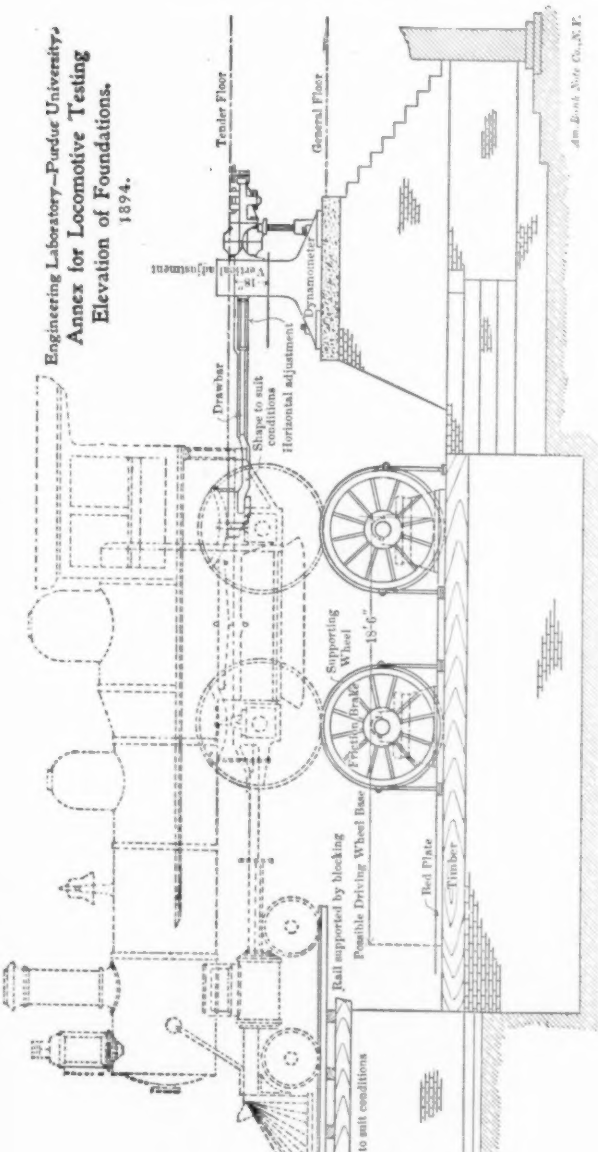


FIG. 408.—ELEVATION OF THE SECOND PURDUE PLANT.

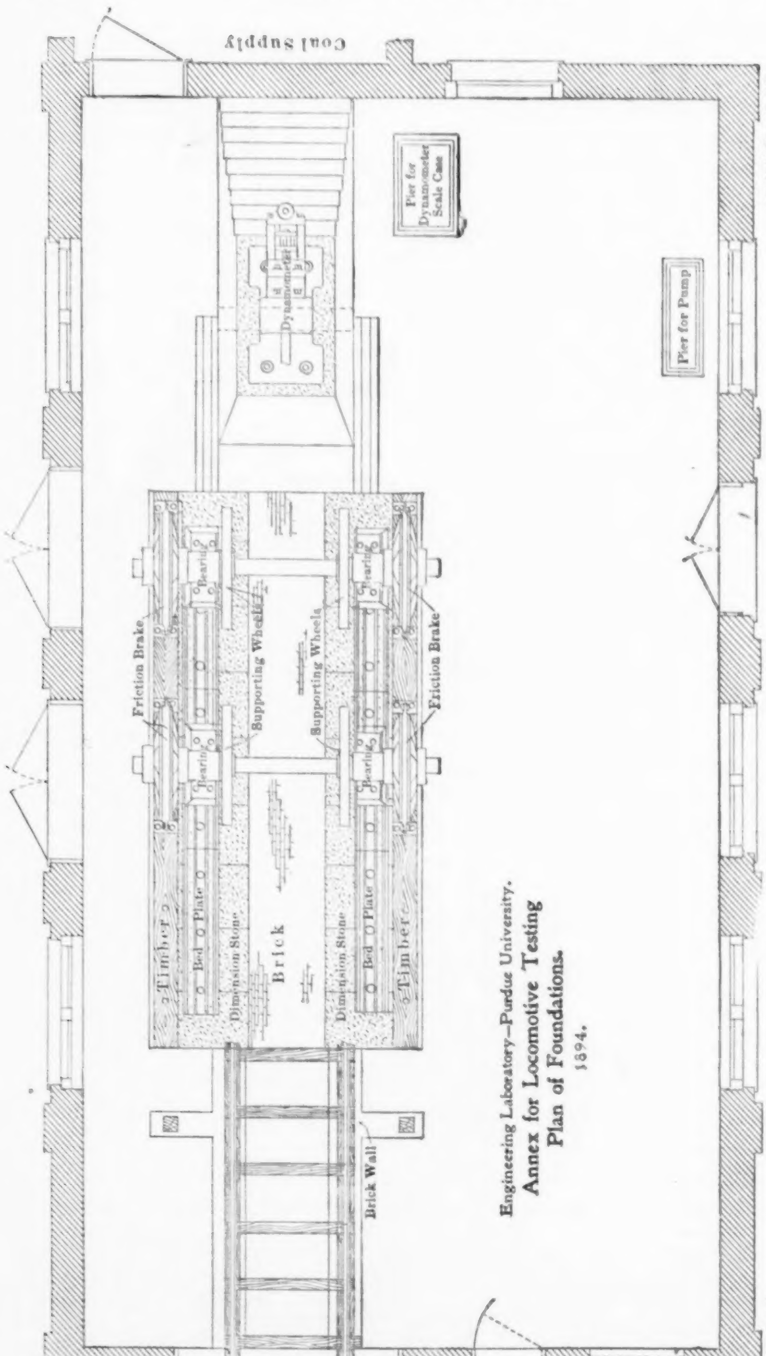


FIG. 409.—PLAN OF SECOND PURDUE PLANT.

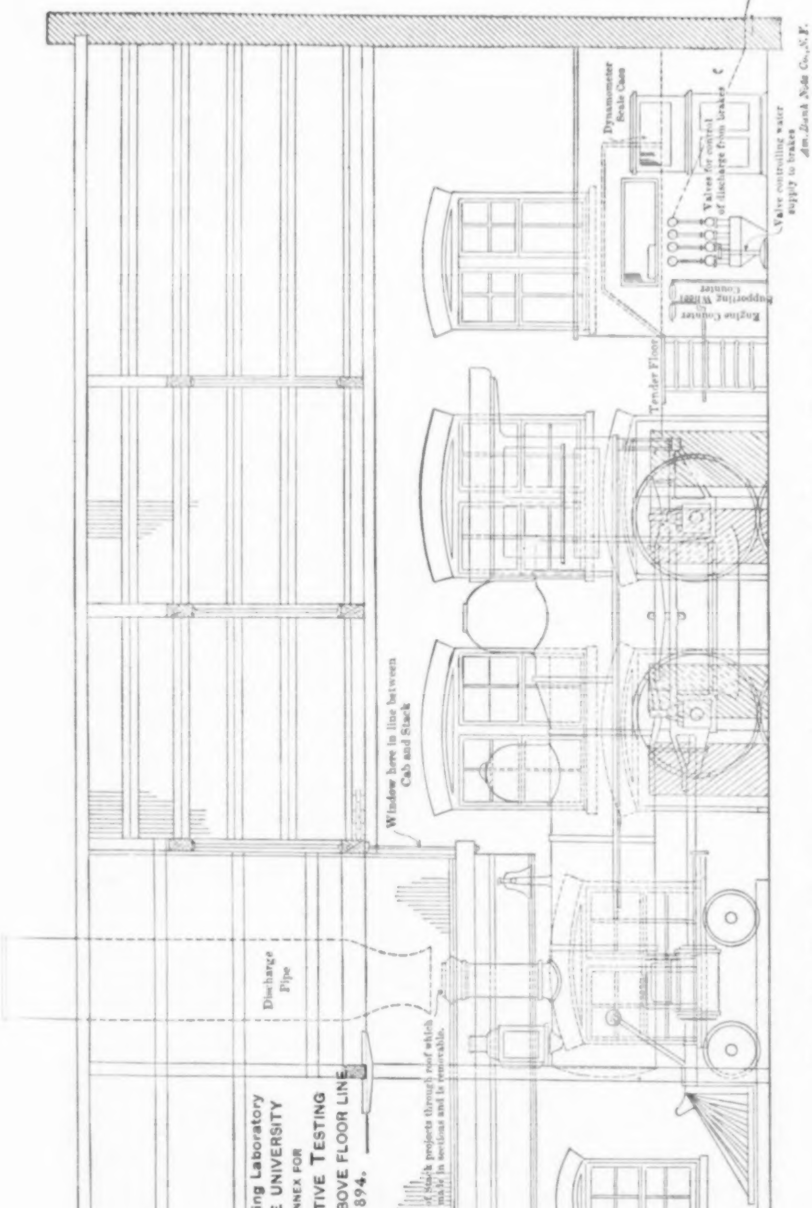


FIG. 410.—ELEVATION SHOWING ACCESSORY APPARATUS, SECOND PURDUE PLANT.

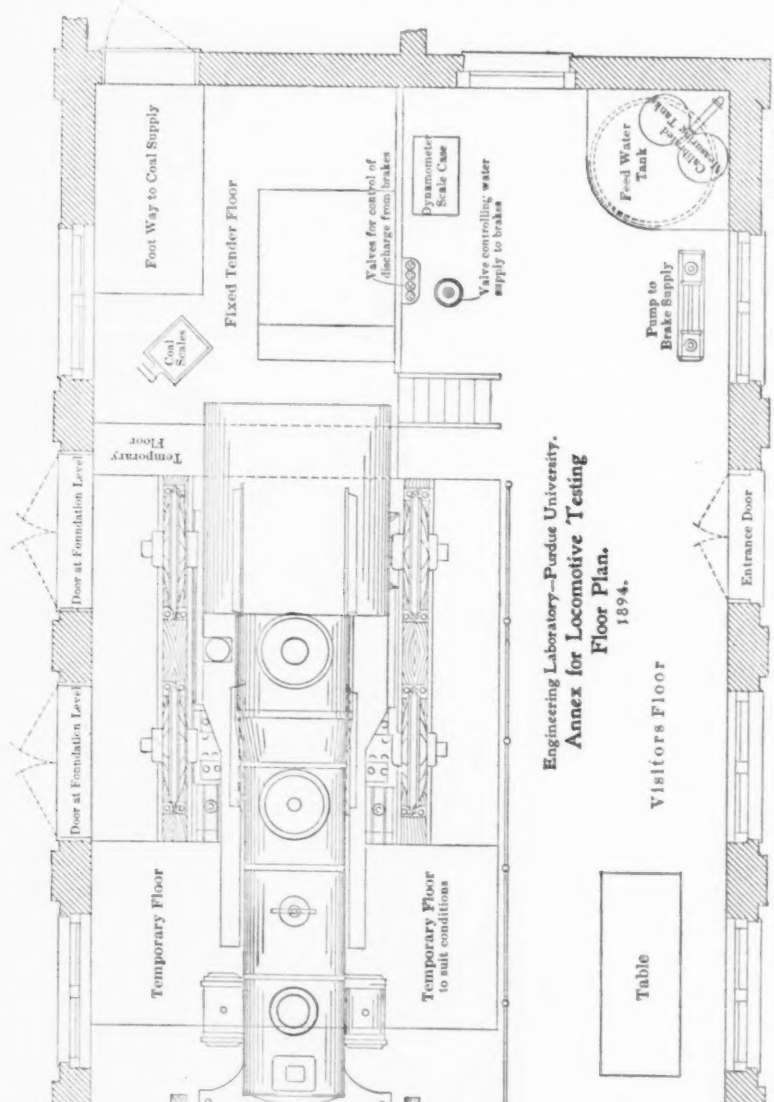


FIG. 411.—ACCESSORY APPARATUS OF SECOND PURDUE PLANT.

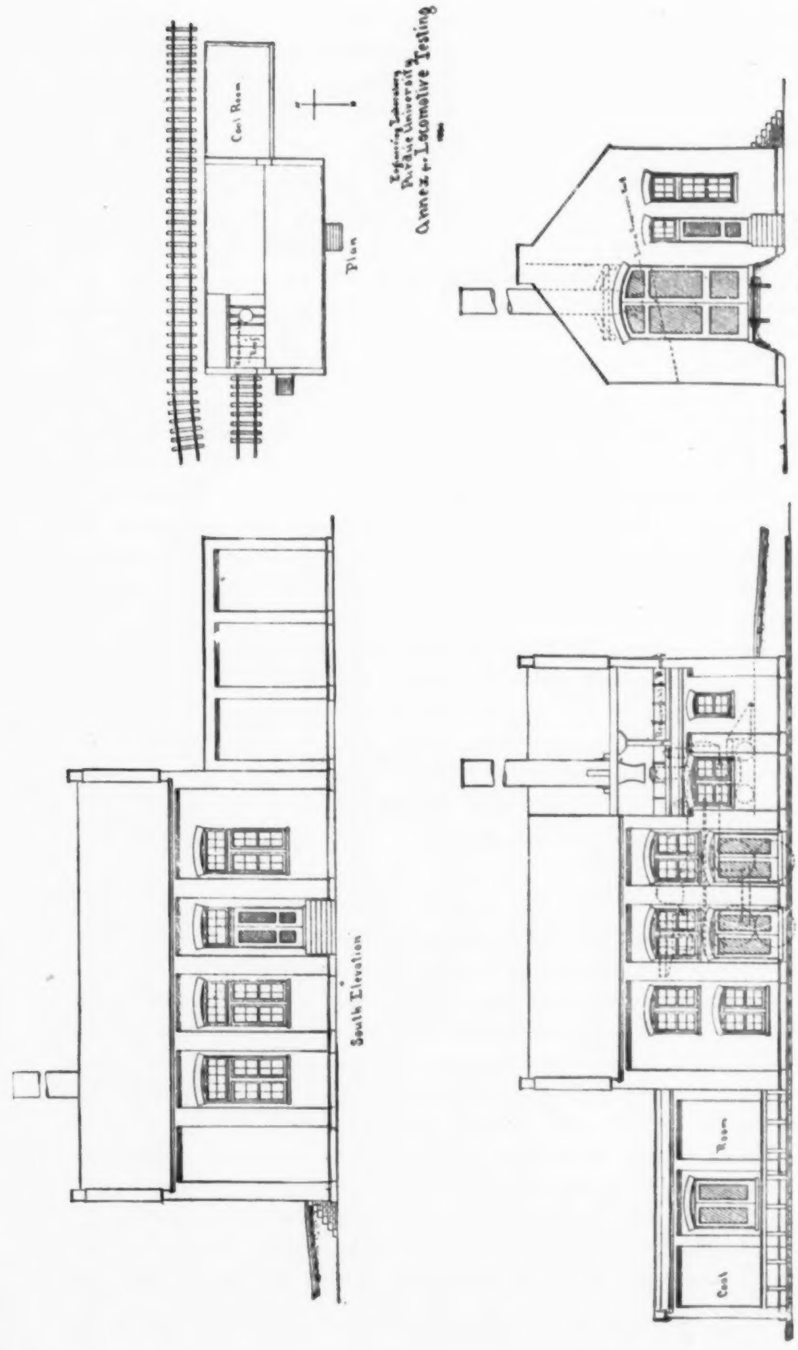
ing for a particular locomotive, the new one is arranged for the accommodation of any locomotive, of which the driving-wheel base does not exceed eighteen and one-half feet.

By reference to Fig. 409, it will be seen that there is provided a wheel foundation of nearly twenty-five feet in length. This is more than sufficient to include the driving-wheel base of any standard eight, ten or twelve-wheeled engine. For engines having six wheels coupled, a third supporting axle will be added to those shown, and for engines having eight wheels coupled, four new axles, having wheels of smaller diameter than those shown, will be used.

The wheel foundation carries cast-iron bed-plates, to which are secured pedestals for the support of the axle boxes. The lower flanges of the pedestals are slotted and the bed-plates have threaded holes spaced along their length. By these means the pedestals may be adjusted to any position along the length of the foundation. The boxes in use at the present time are plain babbited shaft-bearings, and between each bearing and its pedestal is inserted a wooden cushion, now believed to be of no material importance.

The outer edges of the wheel foundations are topped by timbers, to which the brake cases are anchored. The brakes which absorb the power of the engine are those which were used in the original plant. Concerning these, it will be of interest to note that, as used upon the first plant, the oil became very hot about the outside of the case, while that in other parts of the apparatus remained cool. In the re-establishment of the plant, closed circulating pipes were added to the brakes to convey the hot oil from the outer portions of the case back to the centre, the radial curves in the brake-discs serving to maintain a pumping action sufficient to maintain a continuous flow of oil from the outside to the centre of the brake. With these circulating pipes it was found that the open circulation provided in the original design, and described in a previous paper, was unnecessary. In present practice, the brakes are lubricated by a cheap grade of cylinder oil, supplied by the cans shown in Fig. 413. There is always some leakage of oil at the center, but no more than is necessary to insure the lubrication at that point. The drip is caught and returned to the can.

In the original design of the brakes it was assumed that the work absorbed would be proportional to the water pressure exerted upon the copper discs, an assumption which would be true pro-



LOCOMOTIVE LABORATORY.

FIG. 412.—EXTERIOR OF SECOND PURDUE PLANT.

vided the lubricant between the copper plates and the cast-iron moving discs were always in the same condition. In practice it appears that the work is all done on the oil, and any change in the amount of work absorbed, produces a corresponding effect upon the temperature, and hence upon the viscosity of the oil. Other things being equal, therefore, increasing or diminishing the water pressure does not result in a proportional increase or a diminution of the amount of work absorbed. That is, the amount of work absorbed depends quite as much upon the temperature of the oil in the brakes, as upon the pressure exerted by the copper plate upon the moving discs, and, excepting when the speed of rotation is very small, it is found best to use a moderate water pressure, and to vary the load by varying the brake temperature, this being easily accomplished by controlling the amount of circulating water. The water pressure rarely exceeds from four to ten pounds. The pressure between the copper plates and the moving disc being seldom over ten pounds per square inch, there is no perceptible wear of the rubbing surfaces so long as lubrication is maintained. After a total of six million revolutions the copper plates and the cast-iron discs, which constitute the rubbing surfaces of the four brakes, exhibited no trace of wear except in a few places on the copper plates where, owing to imperfections in the surface, small areas have received a concentration of pressure. Since the work absorbed upon the brakes depends largely upon the temperature of the oil between its rubbing surfaces it will appear that the maximum load will be greatest when the heat developed between the rubbing surfaces is most quickly conducted away. The copper plates in the brakes under consideration were made $\frac{3}{16}$ of an inch in thickness. It is probable that less cooling water would be required, and the action of the brakes would be improved if the thickness of these plates were considerably reduced. It is now believed that an amount of metal sufficient only to withstand the shearing forces, to which the action of the brake subjects them, will serve best.

The design of the traction dynamometer was a matter of some difficulty. That which was finally adopted consists of the weighing head of an Emery testing machine, the hydraulic support of which is capable not only of transmitting the stress it receives, but also of withstanding the rapid vibration which the draw-bar transmits to it. The apparatus is of 30,000 pounds capacity, and at the same time so sensitive that one standing in front of the locomo-

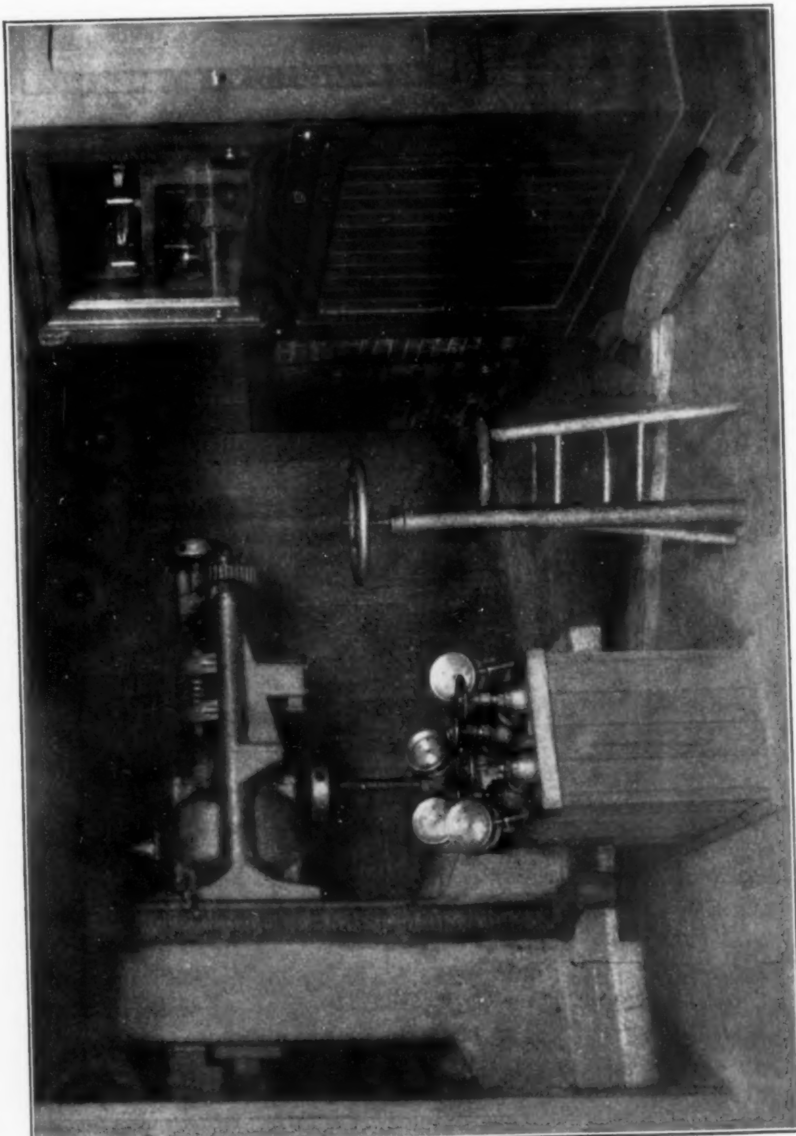


FIG. 413.—LOCOMOTIVE LABORATORY, PURDUE UNIVERSITY.

Dynamometer and Valves Controlling Brakes.

tive may press with his fingers upon its pilot and cause a deflection in the needle of the dynamometer. A massive brick pier, reinforced by steel, forms the foundation of the dynamometer.

In view of the enormous force which a locomotive is capable of exerting it would appear, at first sight, that an error of 50 or even 100 pounds in the determination of draw-bar stresses would be of slight consequence, and that great accuracy in this matter is not required. Under some conditions this conclusion is far from true. The work done at the draw-bar is the product of the force exerted multiplied by the space passed over; if the force exerted is great and the speed small a little error in the draw-bar stress is not a matter of great importance, but if the reverse conditions exist, if the speed is high and the draw-bar stress low then it is absolutely necessary that the draw-bar stress be determined with great accuracy. Moreover, high speeds necessarily involve low draw-bar stresses. Considerations such as these were deemed of sufficient importance to justify the purchase of the most accurate dynamometer whch could be found.

As is well known, the arrangement of the hydraulic support of the Emery testing machine permits the weighing scale to be at any convenient distance from the point where the stresses are received. Figs. 408 and 409 show only the receiving end of the apparatus. The draw-bar connects with this apparatus by a ball joint, which leaves its outer end free to respond to the movement of the locomotive on its springs. A threaded sleeve allows the draw-bar to be lengthened or shortened for a final adjustment of the locomotive to its position upon the supporting wheels, and provision is made for a vertical adjustment of the entire head of the machine upon its frame.

Figs. 410 and 411 show the arrangement of floors. The "visitors' floor" and the fixed floors adjoining are at the level of the rail. The open space over the wheel foundation is of such dimensions as will easily accommodate an engine having a long driving-wheel base, movable or temporary floors being used to fill in about each different engine, as may be found convenient. The temporary flooring is that employed for Purdue's first locomotive.

The level of the "tender floor" is at a sufficient height above the rail to serve as a platform from which to fire. From this a run-way leads to the coal room. At the head of the run-way, set flush with the floor, is a platform scale, which serves for weighing coal as it is delivered to the firemen.

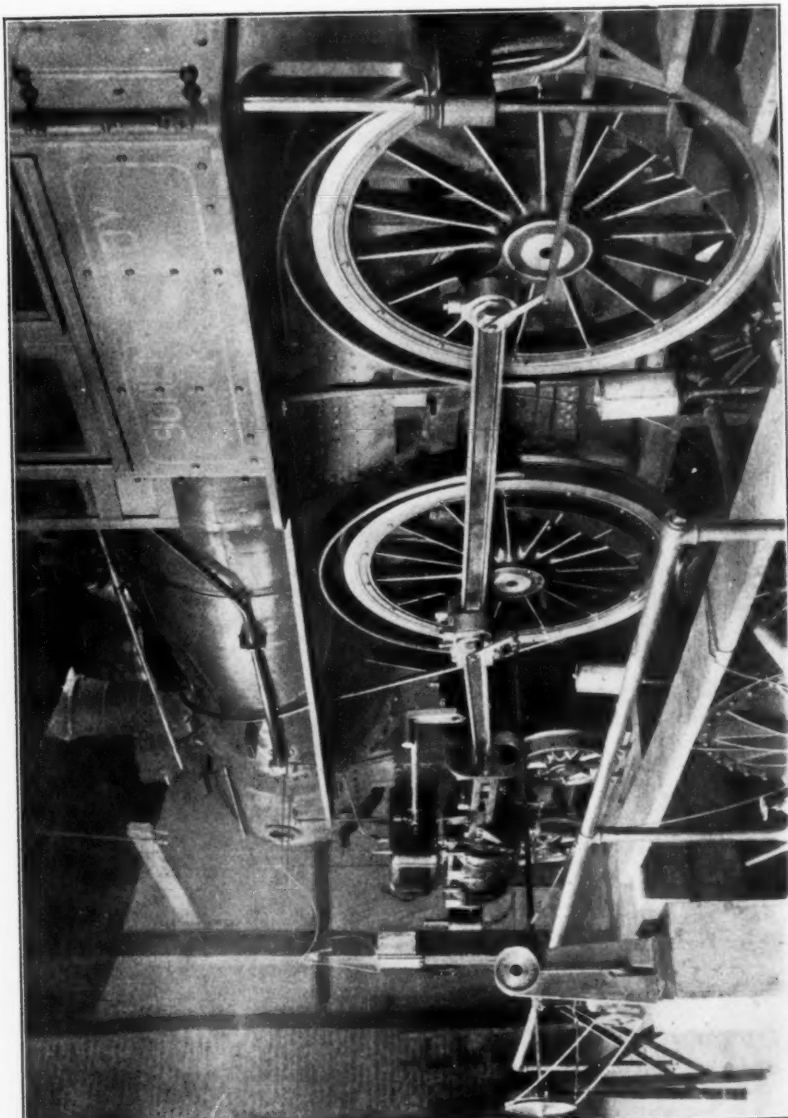


FIG. 414.—LOCOMOTIVE LABORATORY, PURDUE UNIVERSITY.
General View.

The feed-water tank, from which the injectors draw their supply, is shown in the lower right-hand corner of Fig. 411. Above this supply tank are two small calibrated tanks, so arranged that one may be filled while the other is discharging, by use of which the quantity of feed delivered is determined.

The steam pump shown on the visitors' floor is for the purpose of supplying water under pressure to the friction brakes which load the engine.

Fig. 412 presents several views of the locomotive laboratory. The entrance door, which opens upon the visitors' floor, is shown in the south elevation. It is approached from the general laboratory, 75 feet away. The north and west elevations show the roof construction, by which the upper end of the locomotive stack is made to stand outside of the building. The roof sections shown may be entirely removed, and a door in the cross-wall, which extends between the removable roof and the main roof, provides ample height for the admission of the locomotive to the building. A window in this door serves to give the firemen a clear view of the top of the locomotive stack, which at the time the plant was installed was thought to be desirable for good work in firing. Above the stack is a pipe to convey the smoke clear of the building. To meet changes in the location of the stacks of different locomotives, this pipe may be moved to any position along the length of the removable roof.

It appears to be unnecessary to describe in detail the methods of attaching calorimeters, pyrometers, and other minor apparatus, except in so far as these appear in Figs. 413 and 415.

The six years which follow the mounting of the locomotive "Schenectady," upon the original plant of the Purdue Laboratory, were marked by unusual progress in locomotive design, and by the end of that period the experimental engine had ceased to be representative of the most approved practice. For example, steam pressures of 200 pounds per square inch and more, then became common, while the boiler of "Schenectady" had been designed for a maximum pressure of 140 pounds. Moreover, the progress of the work of the laboratory had defined many problems affecting the performance of simple locomotives, and it seemed best to provide for work which would contribute to the solution of that arising from the employment of compound locomotives. Early in 1897, therefore, it was decided that locomotive "Schenectady" should be disposed of and another engine

which would better serve the purposes of the laboratory secured to take its place. The new locomotive, "Schenectady No. 2," arrived at the laboratory in October, 1897, and entered at once upon the work for which it was designed. It is a 16 by 24 eight-wheel engine, of 107,200 pounds total weight, with sixty-nine inch drivers, and a boiler designed for a working steam pressure of 250 pounds. The principal work of the plant during the last seven years has been in connection with this engine.

With reference to the results which have been obtained from the second Purdue plant during the ten years which have elapsed since its establishment in 1894, it is perhaps not too much to say that some of the facts concerning locomotive performance, which to-day are generally accepted, and hence quite commonplace, were first made known, or, if previously surmised, were confirmed by results obtained from this plant. The limitations affecting such simple factors as indicated horse-power and rates of combustion, are now commonly understood, but were much in doubt prior to the advent of the Purdue plant. It has shown that the American locomotive will, under favorable conditions, develop an indicated horse-power upon the consumption of from twenty-five to twenty-six pounds of steam per hour; a fact which ten years ago was accepted with expression of surprise. It has served to define the evaporative performance of the locomotive boiler. It has demonstrated, contrary to a belief once held, that steam from the locomotive boiler is practically dry. In the discussion concerning the advantage of controlling the output of power by means of the reverse-lever, rather than by the throttle, which was current ten years ago, a conspicuous part was given results obtained from the laboratory. It has shown that in a simple engine, the maximum cylinder performance results from a cut-off, which is between the limits of $\frac{1}{2}$ and $\frac{1}{3}$ stroke, and has demonstrated the whole relation of steam consumption to cut-off, under a considerable range of speeds. Among the more specialized problems that were undertaken may be mentioned a study of the value of the steam pipes within the smoke-box when regarded as superheaters; the extent to which fuel loss is involved in the discharge of sparks from the stack, under different rates of combustion; the action of the steam-jet as a means of producing draught, and an investigation of the proportions which for maximum efficiency should be given the exhaust-pipes and stack. Thus far three different locomotives have been on the second plant,

"Schenectady No. 1," the Strong balanced compound, which was thoroughly tested, and "Schenectady No. 2." The plant is concerned at the present time with an exhaustive study, which is planned to disclose the advantage of high steam pressure, in the course of which pressures as high as 250 pounds are being employed.*

IV. *Experiments of Mr. Robert Quayle at Kaukauna.*—In 1893 the American Railway Master Mechanics' Association appointed a committee to investigate and report on exhaust nozzles and steam passages. This committee, appreciating the difficulties to be met in conducting tests upon the road, had arranged to use in their work the testing plant of Purdue University, but before the actual work was begun, the fire of January, 1894, interrupted their plans. Having been thus temporarily deprived of the means for advancing their work, the chairman of the committee, Mr. Robert Quayle, then master mechanic of the Chicago and Northwestern Railway, designed and erected at South Kaukauna, Wis., a temporary testing plant, which served a very useful purpose.†

A four-wheeled truck from a passenger car was lengthened out

---

* The following are among the more formal publications which have been issued since the re-establishment of the Purdue locomotive testing plant:

"Notes Concerning the Performance of Purdue Locomotive Schenectady." Proceedings of the Western Railway Club, May, 1896.

"Steam Pipes Within Locomotive Smoke-boxes as a Means of Superheating." Railway Review, July 23, 1894.

"The Effect of High Rates of Combustion Upon the Efficiency of Locomotive Boilers." Proceedings of the New York Railroad Club, September, 1896.

"Some Factors Affecting the Power of Locomotives." Proceedings of the New England Railroad Club, December, 1901.

"Tests of the Boiler of the Purdue Locomotive." *Transactions* of the American Society of Mechanical Engineers, Volume XXII.

"Recent Progress in the Design of Locomotive Front Ends." Proceedings of Central Railway Club, November, 1903.

"The Form of the Exhaust Jet." Constituting a part of the "Report on Exhaust Pipes and Steam Passages," by a committee. Proceedings of American Railway Master Mechanics' Association, Volume XXIX.

Other publications which have been inspired by the work of the laboratory, and which present the results obtained therefrom are as follows:

"Locomotive Sparks." W. F. M. Goss. John Wiley & Sons. 1902.

Report of committee on "Efficiency of High Steam-Pressure for Locomotives." Proceedings American Railway Master Mechanics' Association, 1898, Volume XXXI.

† For description and plans see "Proceedings American Railway Master Mechanics' Association," for 1894.

to make the spacing of its wheels agree with that of the drivers of the locomotive it was proposed to mount. The truck was then turned bottom side up and securely fastened to a heavy timber foundation, sunk in a pit, to bring the tops of the wheels level with the rails of a nearby track. The truck thus mounted was

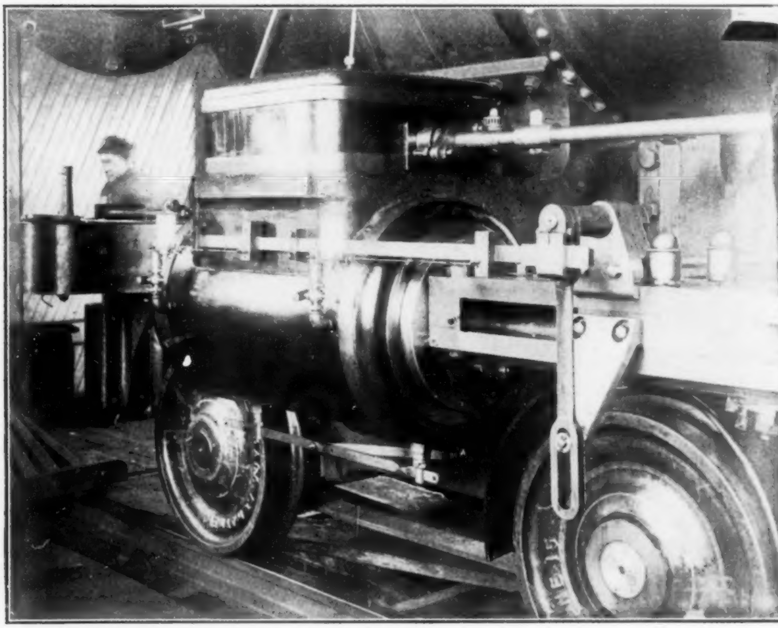


FIG. 415.—INDICATOR RIGGING, SECOND PURDUE PLANT.

fitted with its usual axle boxes and brasses, but the flanges of the wheels were removed. Each of the four supporting wheels, which were thus provided, was fitted with two brake shoes, which were brought into action by means of levers and an air cylinder of the usual form, but to make easier the maintenance of the braking action, water instead of air was used in the brake cylinder. Jets of water were arranged to play on the brake shoes and wheels for the purpose of carrying away the heat developed. Upon the mechanism thus hastily provided, an eight-wheeled locomotive was mounted, heavily braced at the rear to hold it in position. When in running the vibration became so severe as to endanger its fastenings, a second locomotive was backed down the track

ahead of it, which, with the brakes set, proved an effective stop. Thus mounted, the experimental locomotive was operated under a considerable range of speed and load; its cylinder power was indicated, and the draft and back pressure determined for a series of changes in the proportions of the mechanism of the front end.

The plant served to permit the operation of the locomotive at maximum power and at speeds as high as forty miles per hour. A view of this plant from a photograph is shown in Fig. 416.* The period of its operation was short, but the results obtained from it, have served an important part in the development of correct theory with reference to the action of the front end. Its establishment has served also to demonstrate the comparative ease with which a full-sized locomotive may be mounted for experimental work.

V. *The Plant of the Chicago & Northwestern Railroad Co.*—In 1895, Mr. Quayle, having become superintendent of motive power of the Chicago & Northwestern system, a permanent locomotive testing plant was erected at the company's shops at West Fortieth Street, Chicago. Its purpose was to afford facilities for investigating important questions of design upon a road having approximately a thousand locomotives.†

A side and end elevation of the plant is shown by Fig. 417. It occupies one stall of a busy round-house, an adjacent track being available for a car, or for the tender of the locomotive, from which supplies of fuel are drawn. A platform extending from the car or tender to the foot-board of the locomotive under test, supplies space for weighing the coal as it is wheeled in a barrow across to the firemen. A permanent arrangement of calibrated tanks serves in determining the quantity of water delivered to the locomotive injectors.

The pit, in which the mounting machinery is erected, is fifteen feet wide and twenty-eight feet long. The bottom is covered with gravel eighteen inches deep and drained to the centre. In this gravel, which serves as ballast, are laid heavy cross-ties, on the top of which are heavy longitudinal timbers, capped by continuous bed-plates of cast-iron, *C* (Fig. 418 B). These carry cast-iron journal boxes for the axles of the supporting wheels. The bed-plates have a rack extending throughout their length into which engages

* "Report on Exhaust Nozzles and Steam Passages." Proceedings American Railway Master Mechanics' Association, 1894, page 108.

† "Modern Locomotives." Published by the Railroad Gazette, 1897.

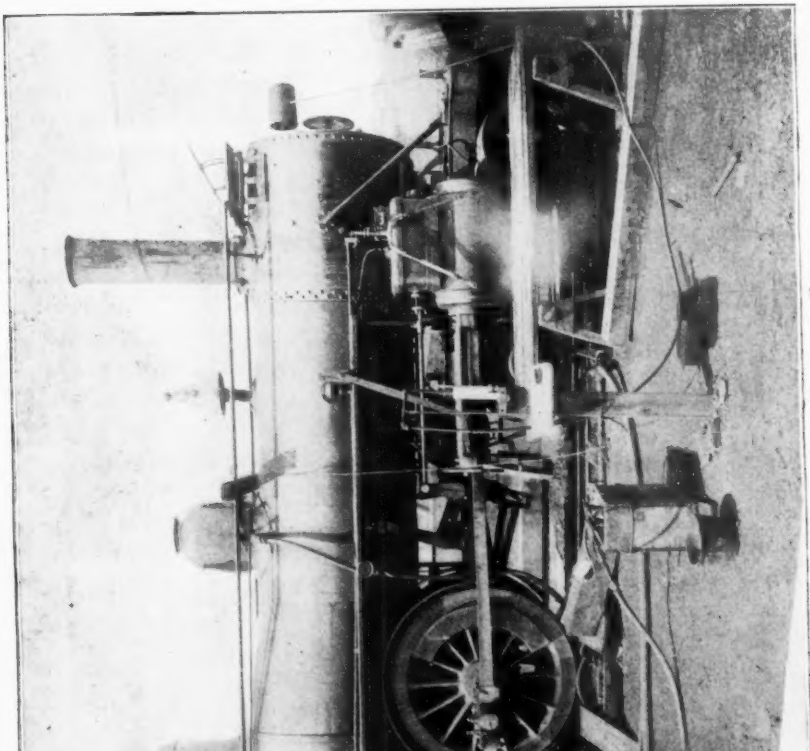


Fig. 416.—First Plant of the Chicago & Northwestern Railway Co.

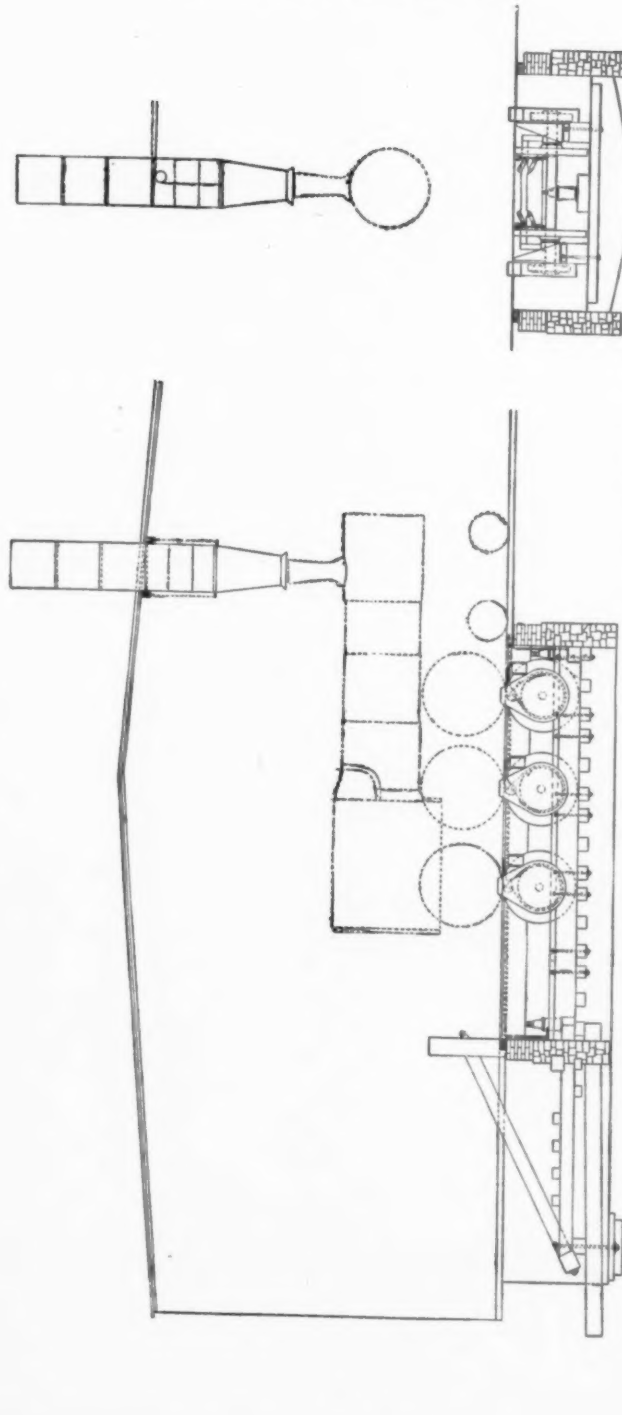


FIG. 417.—GENERAL ARRANGEMENT OF ENGINE TESTING PLANT, C. & N. W. RY.

a small pinion, attached to the journal box. T-slots in the bed-plates receive the bolts of the journal boxes. The whole arrangement is such that by manipulating a ratchet on the shaft of the pinion, the boxes may be easily and quickly moved along the bed-plates to suit the location and spacing of drivers of any locomotive to be mounted upon the plant.

The supporting wheels, *A*, are old fifty-six-inch drivers, with plain steel tires, having their inner edge turned with a radius corresponding to that of a rail head. At present there are but three sets of these wheels, and provision is made for the introduction of a fourth set when needed. The relative arrangement of supporting axles, wheels and brakes is shown in Fig. 417.

The brake-wheels, *B* (Figs. 418), are cast-iron, chilled upon their faces, and thirty-three inches in diameter. About each brake is a steel band, *G*, $\frac{1}{4}$-in. by $5\frac{1}{2}$-ins., the ends of which are attached to the arms of bell-crank lever, *F*. Within the steel band is a series of cast-iron shoes, *D*, which, by a suitable movement of the bell-crank lever may be brought into contact with the brake-wheel. Each shoe has a lug on its back, which projects through a hole in the brake band, and is held in place by a split pin. The longer arm of the lever, *F*, is attached to the piston-rod of an air cylinder, *E*, under the influence of which the shoes are made to bear upon the brake wheel. The trunk pistons of the air cylinders are six inches in diameter and are fitted with cup leathers. The air-cylinders are supplied by a flexible connection, *H*, which permits the supporting axles to be changed along the bed-plate. This leads to a small air-reservoir, which in turn is connected with the air compressor of the shop. The speed of the locomotive on the plant is automatically controlled by means of an ordinary throttling engine-governor in the air-pipe, the pulley of which is belted to one of the supporting axles. When the governor is once set, the brakes will adjust themselves in such a manner as to hold the locomotive at a constant speed. By the use of pulleys of suitable size on the governor, any desired speed of the locomotive may be maintained.

The brakes are cooled by being entirely submerged in water. To permit this the brake wheel and band are surrounded by a sheet-iron casing, *N* (Figs. 418). A packing box is provided to prevent the entrance of water into the journal boxes of the supporting axles, and the slight leakage which occurs at this point gives no trouble. A stream of water passes constantly into the case

through *K* and out through the overflow *I*. Thus far no dynamometer has been applied to this plant, and no attempt is made to measure the draw-bar pull; but the engine is held in position by being coupled to a strongly constructed frame work in the rear. A fixed pipe having vertical adjustment carries away the

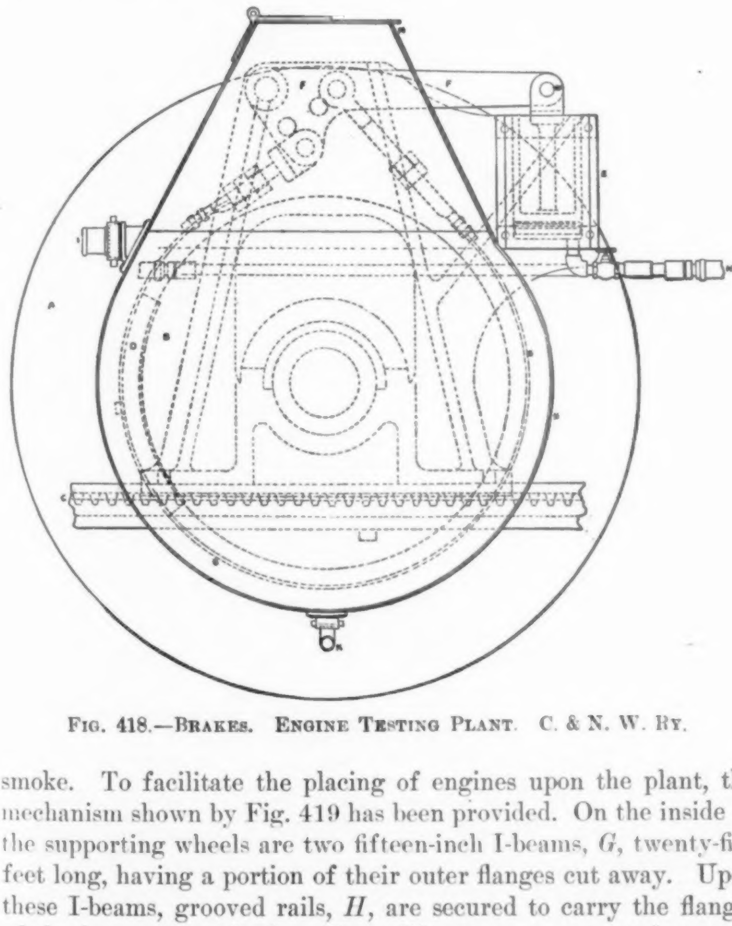


FIG. 418.—BRAKES. ENGINE TESTING PLANT. C. & N. W. RY.

smoke. To facilitate the placing of engines upon the plant, the mechanism shown by Fig. 419 has been provided. On the inside of the supporting wheels are two fifteen-inch I-beams, *G*, twenty-five feet long, having a portion of their outer flanges cut away. Upon these I-beams, grooved rails, *H*, are secured to carry the flanges of the locomotive wheels. These I-beams are supported at each end by a system of link-work, connecting with an air-cylinder, *B*, by means of which they may be raised or lowered as occasion may require. In Fig. 419 the I-beams are shown in their lowest position. By admission of air to the cylinder, the bar, *C*, is raised, elevating the I-beam, *G*, which are guided outward by links, *E*, into

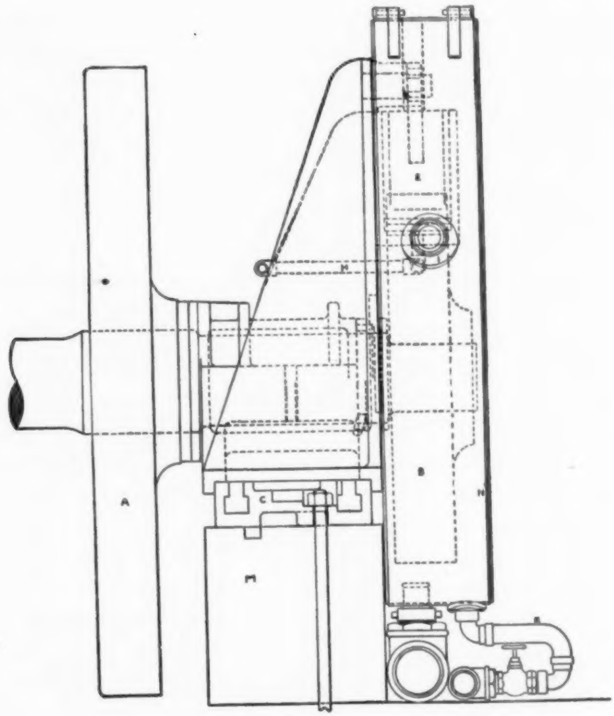


FIG. 418A.

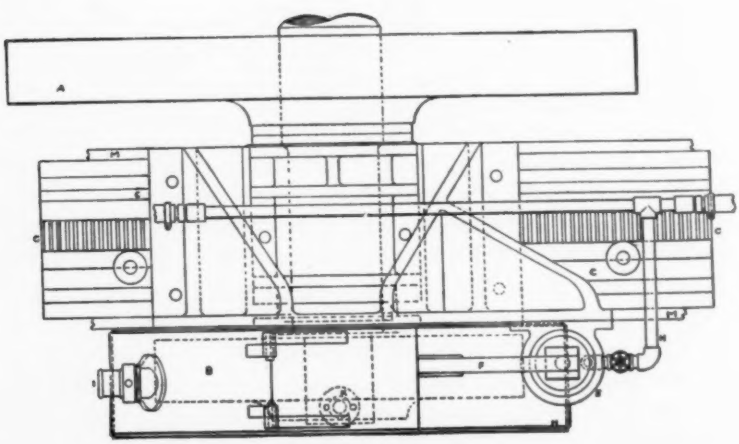


FIG. 418B.

position against the sides of the supporting wheels. Oak blocks are then inserted between the I-beams and the axles to carry the load which may be imposed upon the I-beams. A locomotive, which is to be mounted on the testing plant, is carried by the

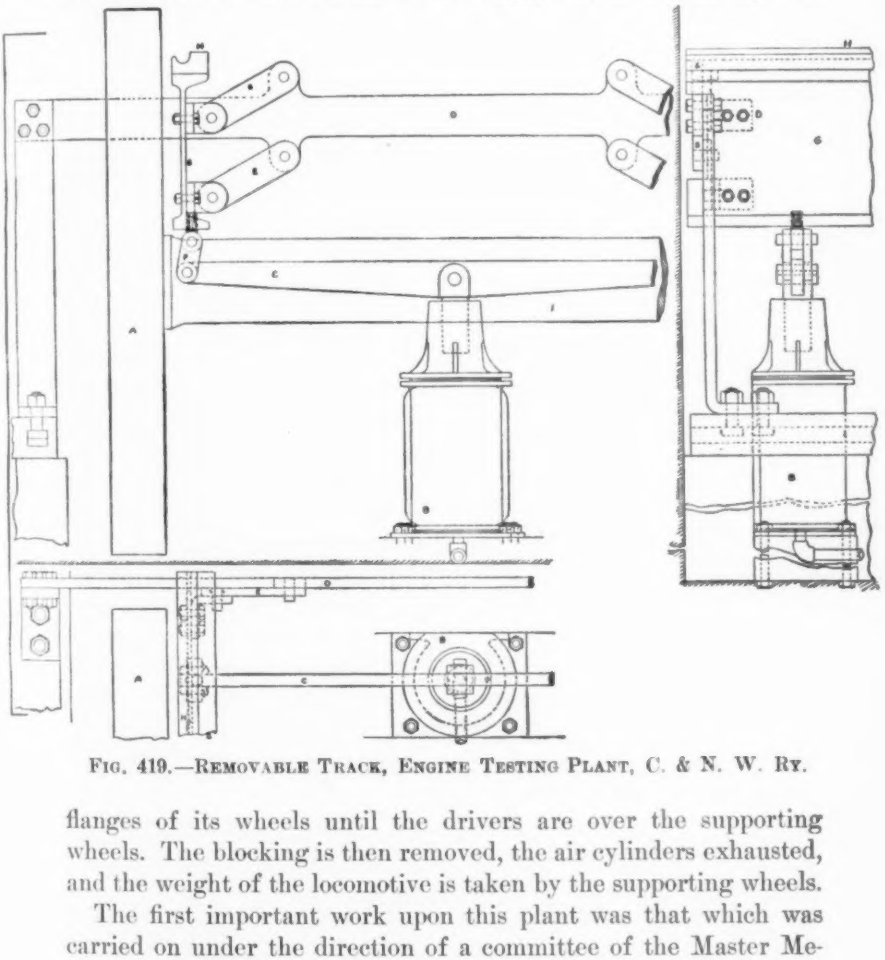


FIG. 419.—REMOVABLE TRACK, ENGINE TESTING PLANT, C. & N. W. RY.

flanges of its wheels until the drivers are over the supporting wheels. The blocking is then removed, the air cylinders exhausted, and the weight of the locomotive is taken by the supporting wheels.

The first important work upon this plant was that which was carried on under the direction of a committee of the Master Mechanics' Association, having in charge an investigation of exhaust pipes and steam passages. An account of this elaborate series of tests is contained in a final report of this committee, published in 1896.* The plant has been used also in determining the relative

* "Report of Committee on Exhaust Pipes and Steam Passages." Proceedings of American Railway Master Mechanics Association, 1896, Volume XXIX.

values of different coals; the efficiency of different types of locomotives at different speeds and cut-offs; the effect upon performance of changes in the lead and inside clearance of values, and in the detection and correction of certain defects in the design of valve-gears. While comparatively few results have been published from this plant, it has proven to be a valuable shop appliance.

VI. *The Plant of the Columbia University.*—In 1899, the Baldwin Locomotive Works presented to Columbia University the locomotive "Columbia," which had been exhibited by them at the Columbian Exposition in Chicago, in 1893, on condition that the University suitably mount it. This engine is a Vauclain compound, having cylinders 13 and 22 by 26 inches, and was designed for fast passenger service. The boiler carries 180 pounds of steam pressure. Its weight on drivers is 83,140 pounds, and its total weight in working order is 126,600 pounds. The general arrangement of the plant is shown by Figs. 420 and 421, the details of the dynamometers by Fig. 422, and the brake construction by Fig. 423. In negotiations leading up to the gift, the University was represented by Prof. F. R. Hutton, under whose direction the machine was finally installed.

The location of the University at Morningside Heights, remote from railway connection, made it necessary to dismantle the engine in anticipation of its delivery. It was then transferred by boat and truck and afterwards re-erected in the laboratory.

Referring first to Figs. 420 and 421, the supporting wheels, *A*, have cast centres and steel tires and are sixty inches in diameter. The bearings, carrying the supporting axles, rest upon timbers, *D*, which cap a steel girder, *E*, extending the full length of the pit. These girders are carried by the end walls of the pit; they are not supported at intermediate points. Motion of the locomotive as a whole, is prevented by its being coupled to a buffing post, *C*. The truck wheels are carried by short sections of track, at the level of the floor. The front and back edges of the pit are capped with iron plates, on which rest I-beams, *F*, extending the length of the pit, carrying rails. These serve as guard rails and may also serve as a means of running the locomotive on or off the plant. The power is absorbed by four Alden brakes, each of 400 nominal horse-power. They are of the multiple disc type, having the form shown in detail by Fig. 423. In this figure, letters *A* indicate the moving discs, and the letters *B* the copper

COLUMBIA UNIVERSITY

LOCOMOTIVE LABORATORY

ELEVATION

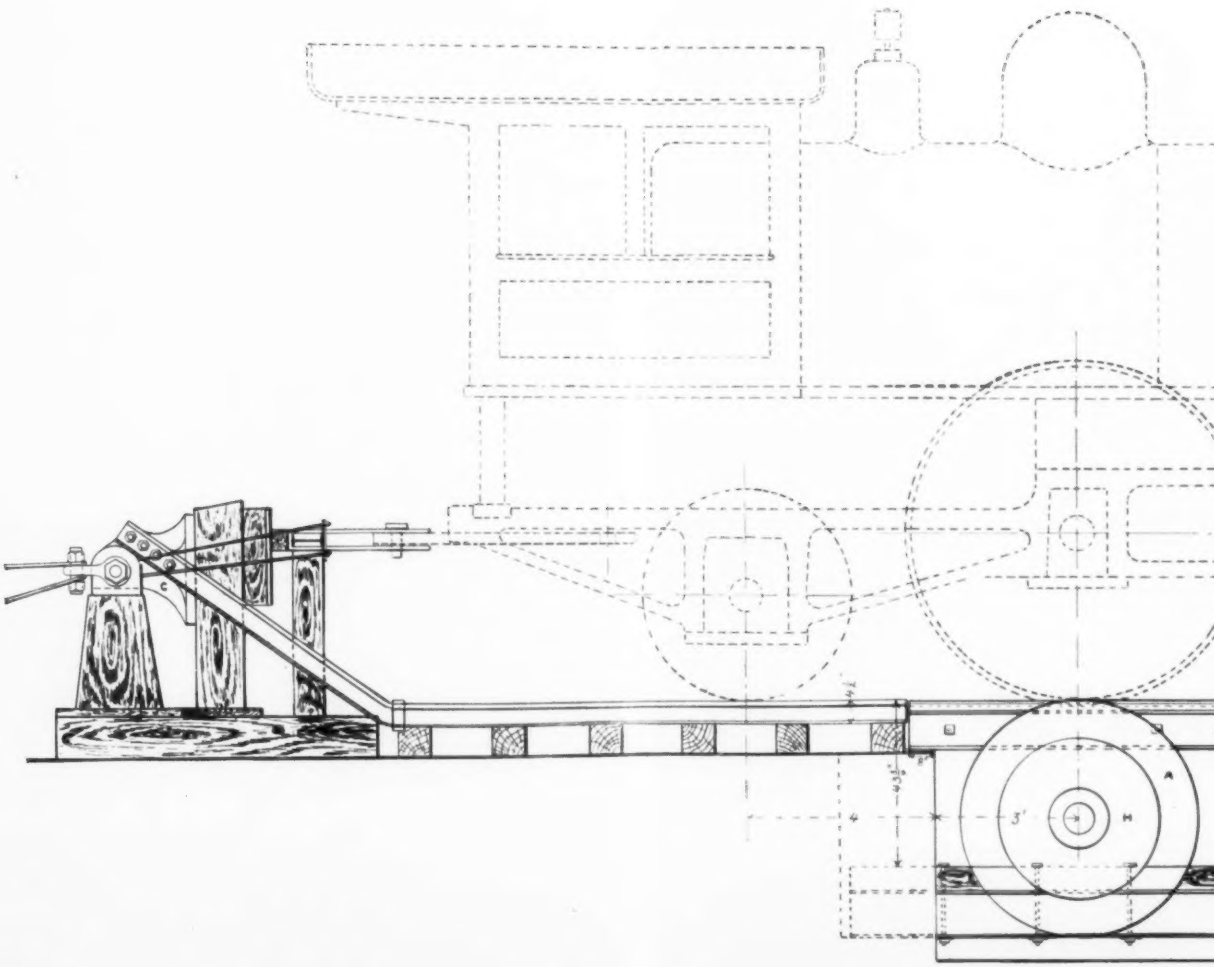


FIG.

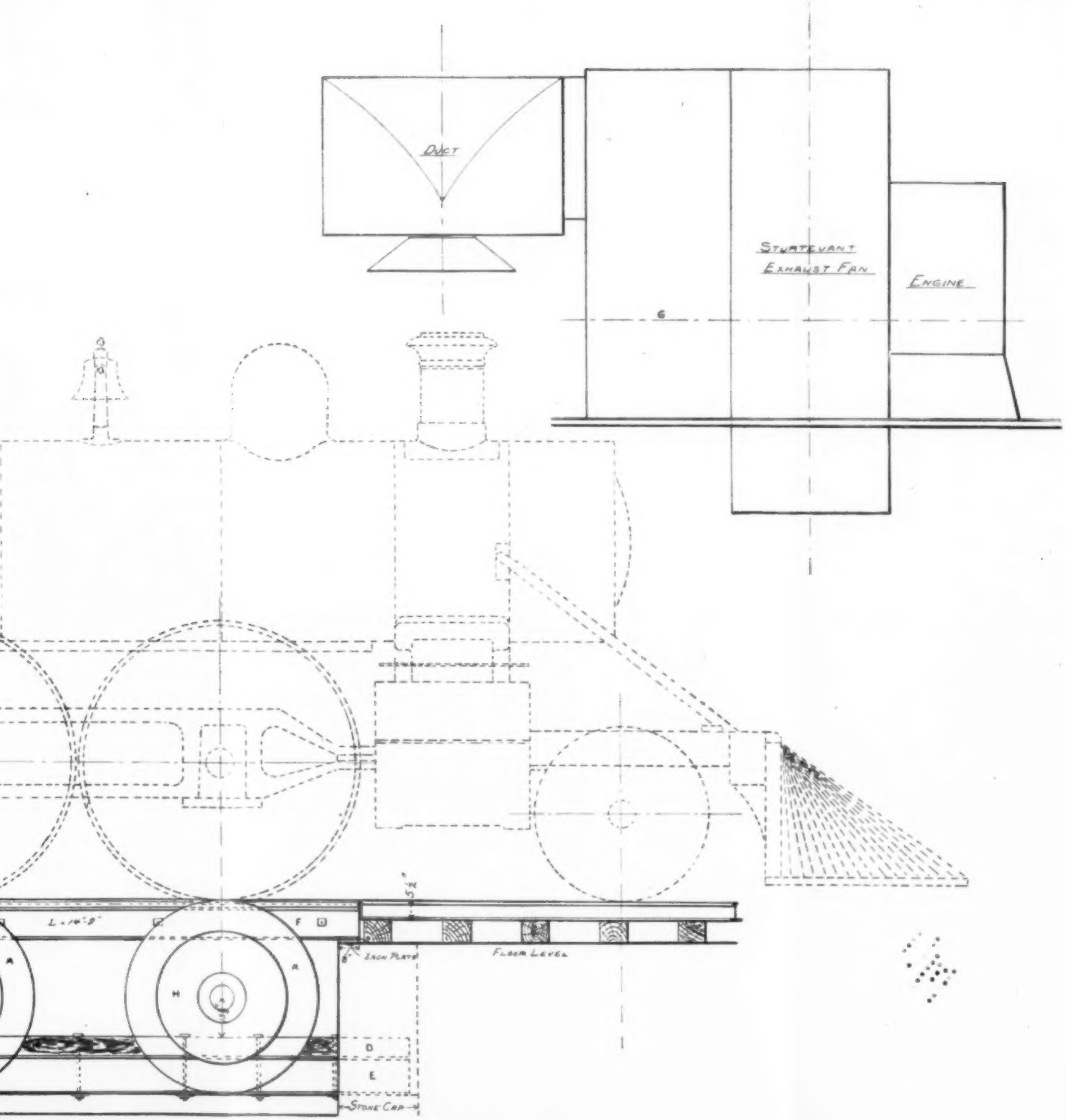
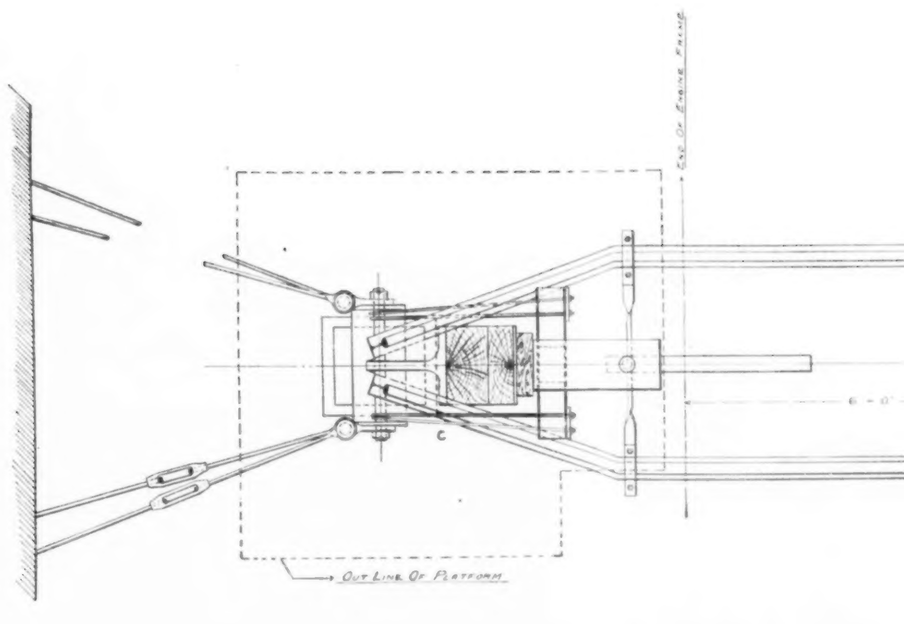


Fig. 420.

COLUMBIA UNIVERSITY
LOCOMOTIVE LABORATORY
PLAN



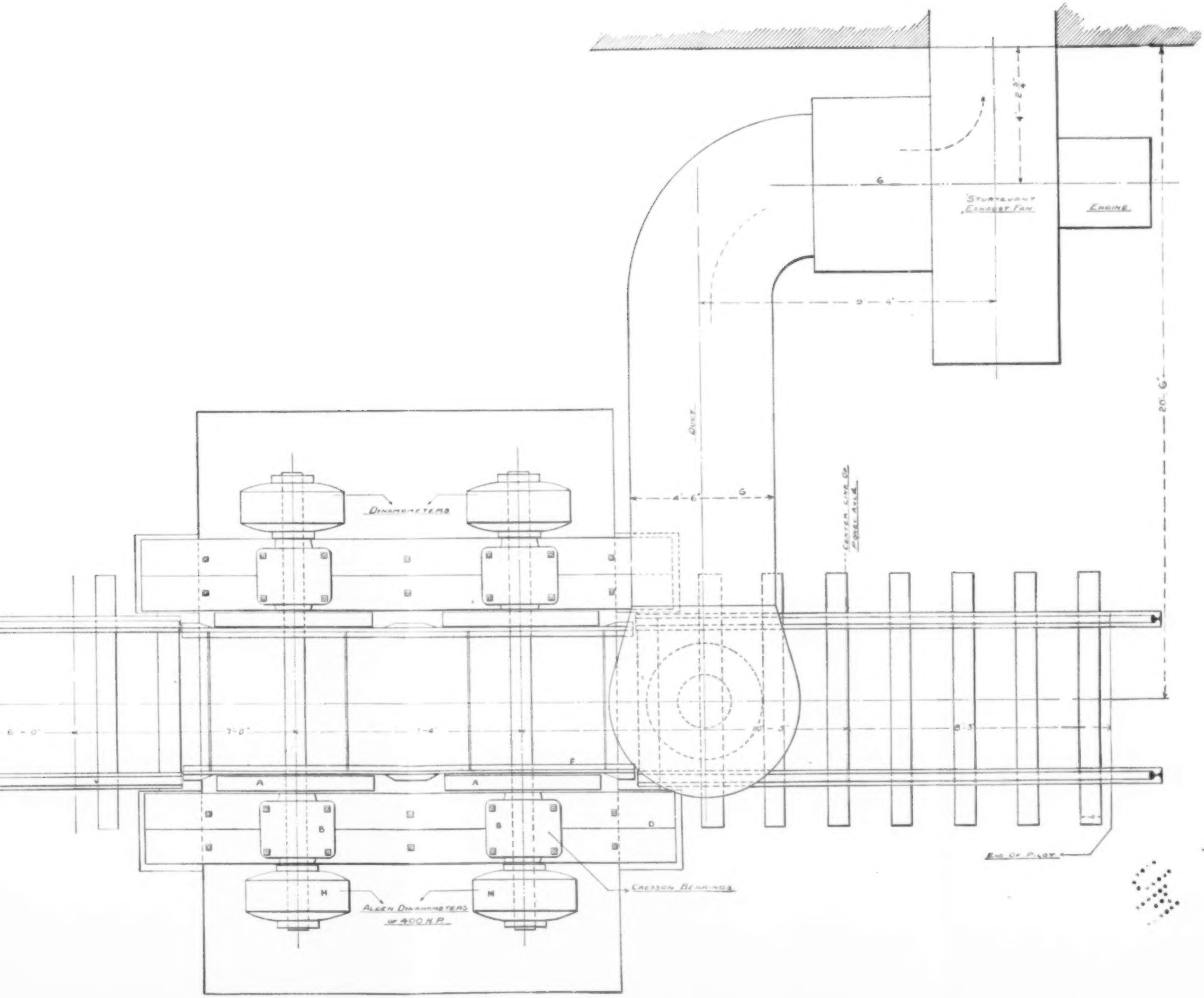

Sturtevant Exhaust Fan
Engine
Duct
Dynamometers
Alden Dynamometers of 400 H.P.
Cresson Bearings
End of Pilot
Center Line of Pivot Axle
9'-4"
20'-6"
4'-6"
7'-4"

Fig. 421.

plates which lie on either side of the discs and which are attached to the enveloping case. Cooling water circulates in the spaces *C*, entering from both sides of the case by the pipe *D* and being discharged from the centre by the pipe *E*. All

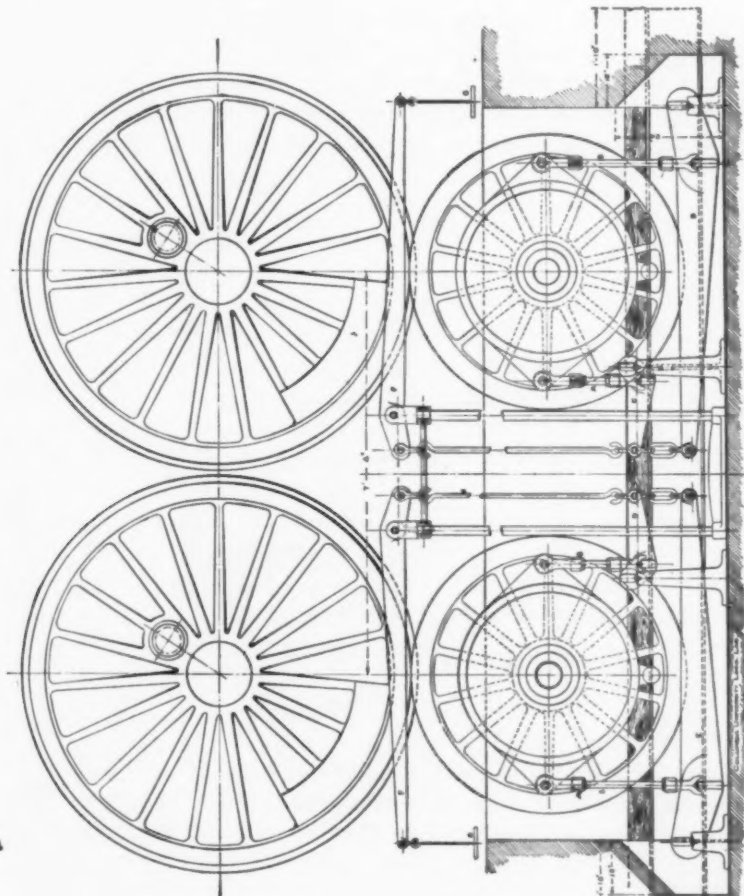


FIG. 422.—ARRANGEMENT OF BRAKES' COLUMBIA UNIVERSITY LOCOMOTIVE LABORATORY.

space between the copper plates and the moving discs is filled with oil.

In operation, the discs *A* turn with the supporting axles while the copper plates *B*, with the case, are prevented from turning by a suitable system of levers. The work done by the brakes is

regulated by varying the water pressure, which controls the pressure of contact of the copper plates with the moving discs. *F* is an automatic regulating valve, designed to control the supply of water to the friction brakes, for the purpose of keeping the dynamometer levers always in balance.

The power absorbed by each of the four brakes is measured by a system of weighing levers, Figs. 422. In this figure, thrust rods, *A* and *B*, connect the brake cases with the levers in such a way that when the engine is in a forward motion, the rods *A* transmit the force downward to the levers *C*, rods *B* transmitting no

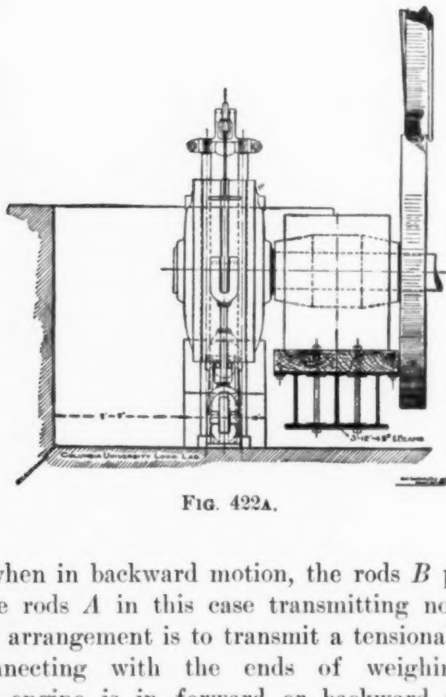


FIG. 422A.

stress; and when in backward motion, the rods *B* push down on levers *D*, the rods *A* in this case transmitting no stress. The effect of this arrangement is to transmit a tensional stress to the rods *E* connecting with the ends of weighing levers *F*, whether the engine is in forward or backward motion. The long arms of the levers *F* carry the weight-holders *G*. Obviously the arms of the levers *C* and *D* are in the same ratio to each other. The power absorbed by the four brakes plus the power absorbed by the journals of the supporting axles is the power delivered from the locomotive drivers. Consequently, the weighings made at the brake, when suitably corrected, may be reduced to an equivalent draw-bar pull. No traction dynamom-

eter, for the direct measurement of the draw-pull is included in the equipment of the plant.

The discharge from the locomotive stack is into a hood connected with a horizontal duct, which is served by a steam-driven Sturtevant blower, delivering into a flue in the wall of the building.

The locomotive plant is fully equipped with accessory apparatus for obtaining all data necessary for complete engine and boiler tests. In addition to the provisions made for normal running, the locomotive has pipe connection with the general steam supply of the laboratory and may be operated by steam from the stationary boilers.

The Columbia plant has served a useful purpose in the routine of a busy laboratory.

VII. *The Plant of the Pennsylvania Railroad System at St. Louis, Mo.**—The Pennsylvania Railroad System has this year installed as a portion of its exhibit in the Department of Transportation at the Louisiana Purchase Exposition at St. Louis, a locomotive testing plant, which is being operated during the seven months of the Exposition. At the close of the Exposition the plant will be permanently located at Altoona as a part of the Company's physical laboratory equipment.

The general arrangement of the plant is shown by Figs. 424, 425, and 426, in which similar letters designate the same parts on all plates. The principles underlying its design are the same as those which have controlled in the development of previous plants; but it is as gratifying as it is logical that this latest development of the testing-plant idea is of greater capacity and far more perfect in its details than any which have preceded it.

A continuous concrete foundation carries the two longitudinal bed-plates, *A*, and extends under and forms a part of the base supporting the traction dynamometer. The bed-plates are provided with longitudinal T-slots, so that the pedestals carrying the

* The Pennsylvania System has thus far published only a general description of their plant (Bulletin No. 2), and as the writer is not authorized to anticipate the company in its plans, a description of its many interesting details is necessarily omitted. The high value of the plant may, however, be judged from the fact that its design has been made to include everything which gave promise of usefulness, either by increasing the facility with which work may be done, or by insuring a higher degree of accuracy in the observed data obtained. The general design with its machinery details was developed by Mr. A. S. Vogt, Mechanical Engineer of the Pennsylvania Railroad Company. Many of its devices are beautiful in their conception and outline.

axles of the supporting wheels can be adjusted to the spacing of the locomotive driving wheels. Two sets of supporting wheels are supplied, one consisting of three pairs, 72 inches in diameter, for use under passenger locomotives; and the other set of five pairs, 50 inches in diameter, for use under freight locomotives. Each supporting-axle has its own pedestal and journal boxes.

The supporting wheels resemble in form, the usual locomotive driving-wheels, having cast-steel centers with tires held by shrinkage and by retaining rings. The contour of the tire is such as to provide a tread similar in form with that presented by the head of a rail; outside of which and separated from it by a wide groove is a light flange designed to keep the tread free from oil which may drip from the locomotive while running.

The equipment includes eight Alden brakes, any of which may be used upon any of the supporting axles, the arrangement being such that one or two brakes may be attached to an axle as conditions may require. All brakes are of the same dimensions and are of the two-disc type.

An electric crane of 10 tons capacity serves the entire space occupied by the plant. By its use the supporting axles with their wheels and pedestals may be easily moved about as conditions may require.

Means for bringing the locomotive safely to its position on the plant form an important part of the installation. The supporting wheels having been placed in positions corresponding to the spacing of the drivers, I-beams resting on the supporting shafts, and extending the full length of the pit, are bolted securely to their inside faces. A grooved rail on the upper flange of the I-beams supports and guides the locomotive, which rolls to place on the flanges of its wheels, leaving the treads free to take their position on the supporting wheels. When the locomotive is in place, the special rails and I-beams are disconnected from the supporting wheels and removed, so as not to interfere with the operation of the plant. In cases where locomotives have flangeless drivers the groove of the rail is filled by a suitable section of rolled steel. Power for moving locomotives on and off the plant is supplied by the overhead crane acting through a suitable rope tackle. The traction dynamometer *G* which measures the draw-bar pull and which has a capacity of 80,000 pounds is a lever machine of the Emery type. Flexible steel fulcrum plates take the place of the more usual knife edges. The weight

of each lever is taken by a vertical plate in a plane intersecting that of the receiving fulcrum plates at their centre of rotation, thus relieving the fulcrum plates of all transverse load. The yoke embracing the dynamometer and to which the draw-bar is at-

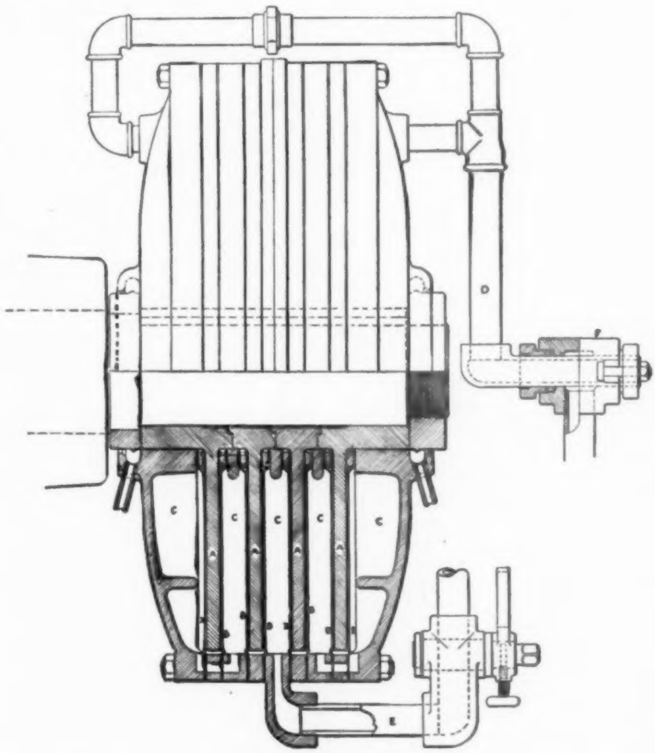


FIG. 423.—COLUMBIA UNIVERSITY LOCOMOTIVE LABORATORY. DETAILS OF ALDEN ABSORPTION DYNAMOMETER.

tached is also mounted on flexible plates and braced by long and flexible rods, to insure frictionless motion in the horizontal plane only. The total motion of this yoke and draw-bar, due to the leverage of the machine and to stress of parts when under full load, does not exceed four one-hundredths of an inch, so that a locomotive exerting a draw-bar pull equal to the full capacity of the dynamometer, will not move forward on the supporting wheels more than the amount specified. The draw-bar is provided with a ball joint, to allow for any side motion of the locomotive on

its springs. Near the base of the dynamometer, the oscillating motion of the ends of the last levers is transformed into a rotary motion by means of steel belts wrapped around a drum and kept in constant tension by suitable clamping devices. The belt drum is mounted on a tube guided in ball bearings, and inside of it

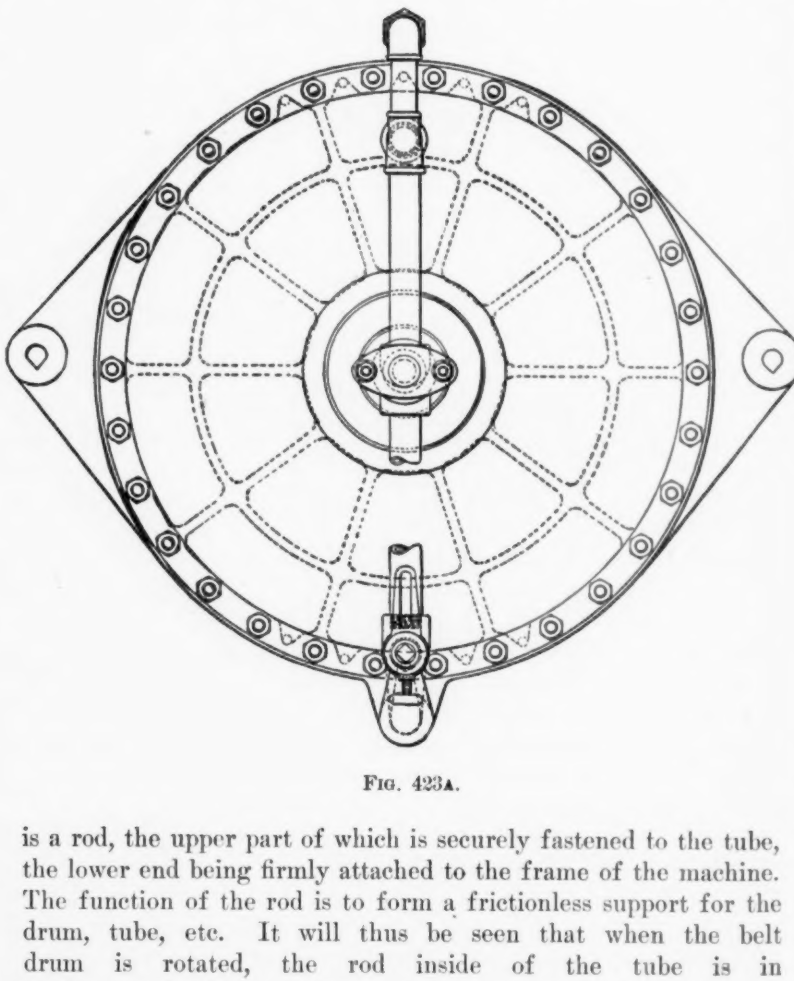


FIG. 423A.

is a rod, the upper part of which is securely fastened to the tube, the lower end being firmly attached to the frame of the machine. The function of the rod is to form a frictionless support for the drum, tube, etc. It will thus be seen that when the belt drum is rotated, the rod inside of the tube is in torsion, and this resistance forms part of the total resistance of the machine, and is a constant for a given travel of the recording pen. To the upper end of the tube already mentioned are secured two radial arms, the extremities of which

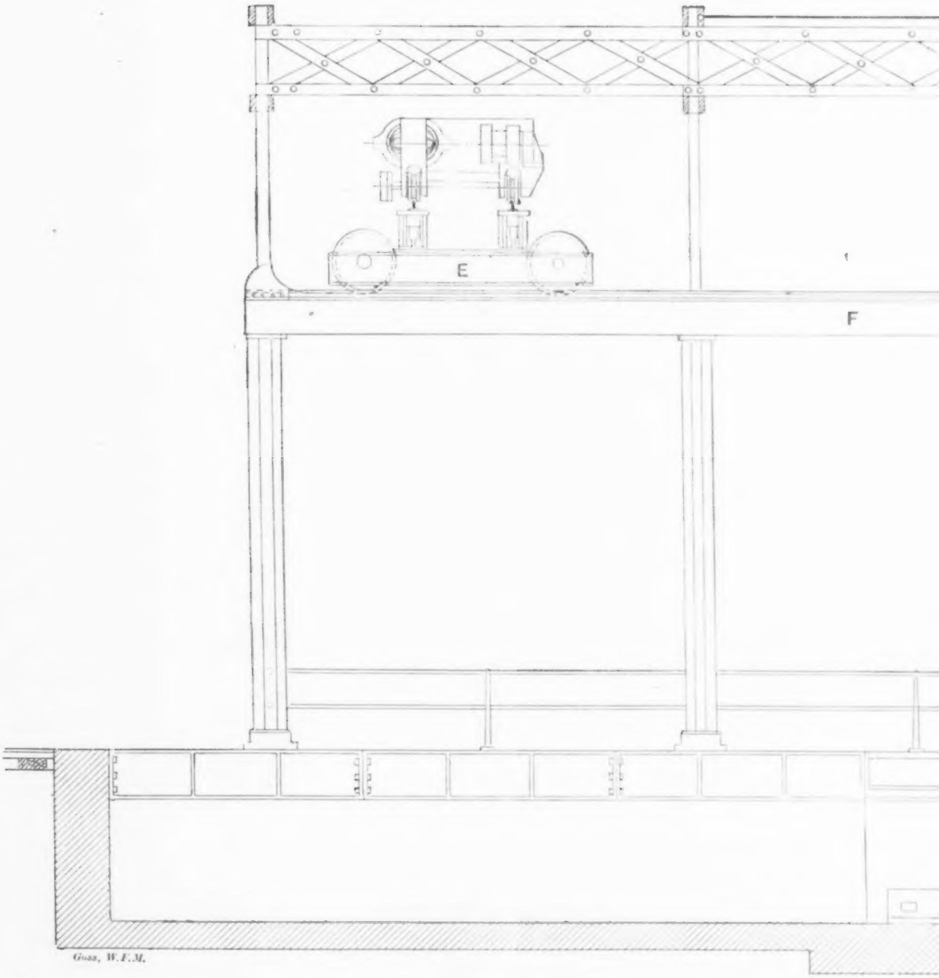
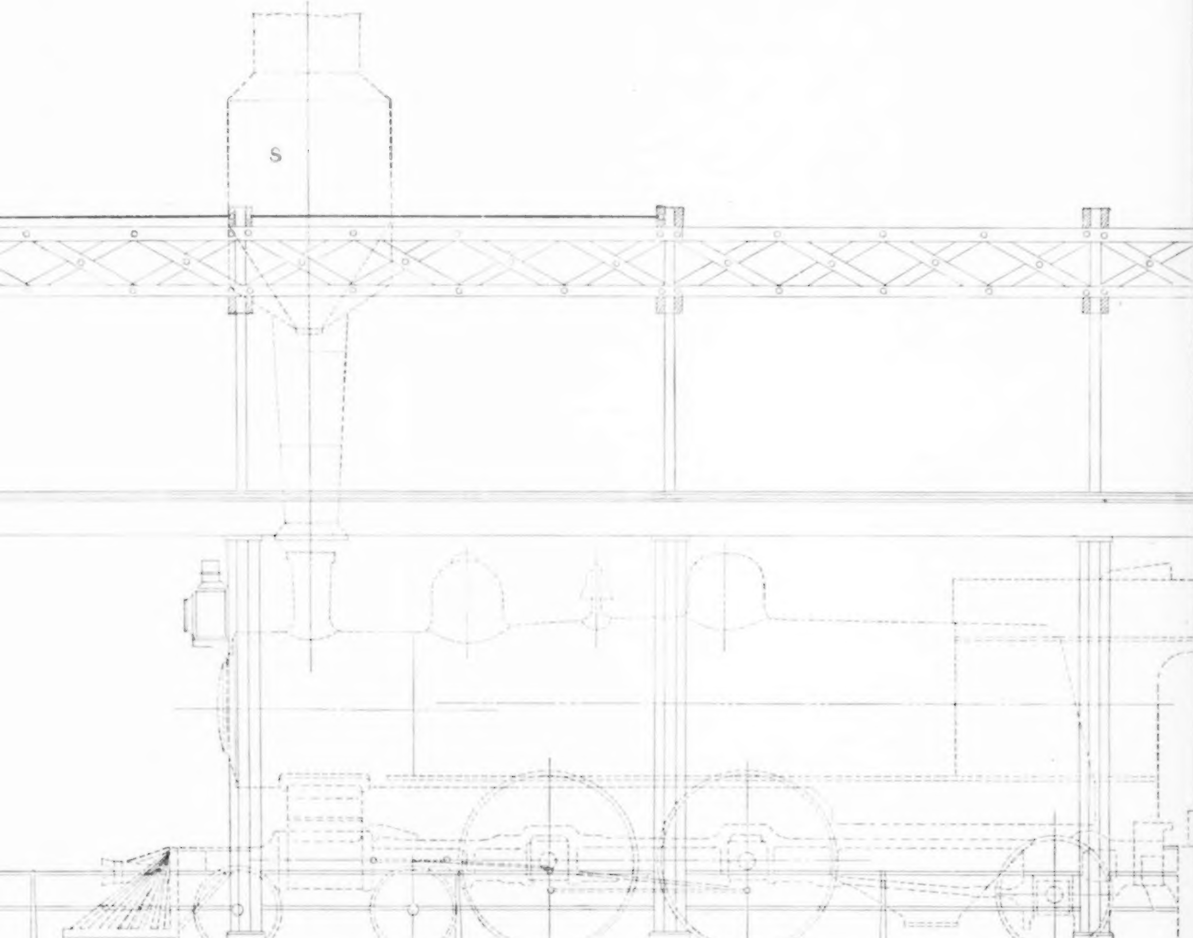
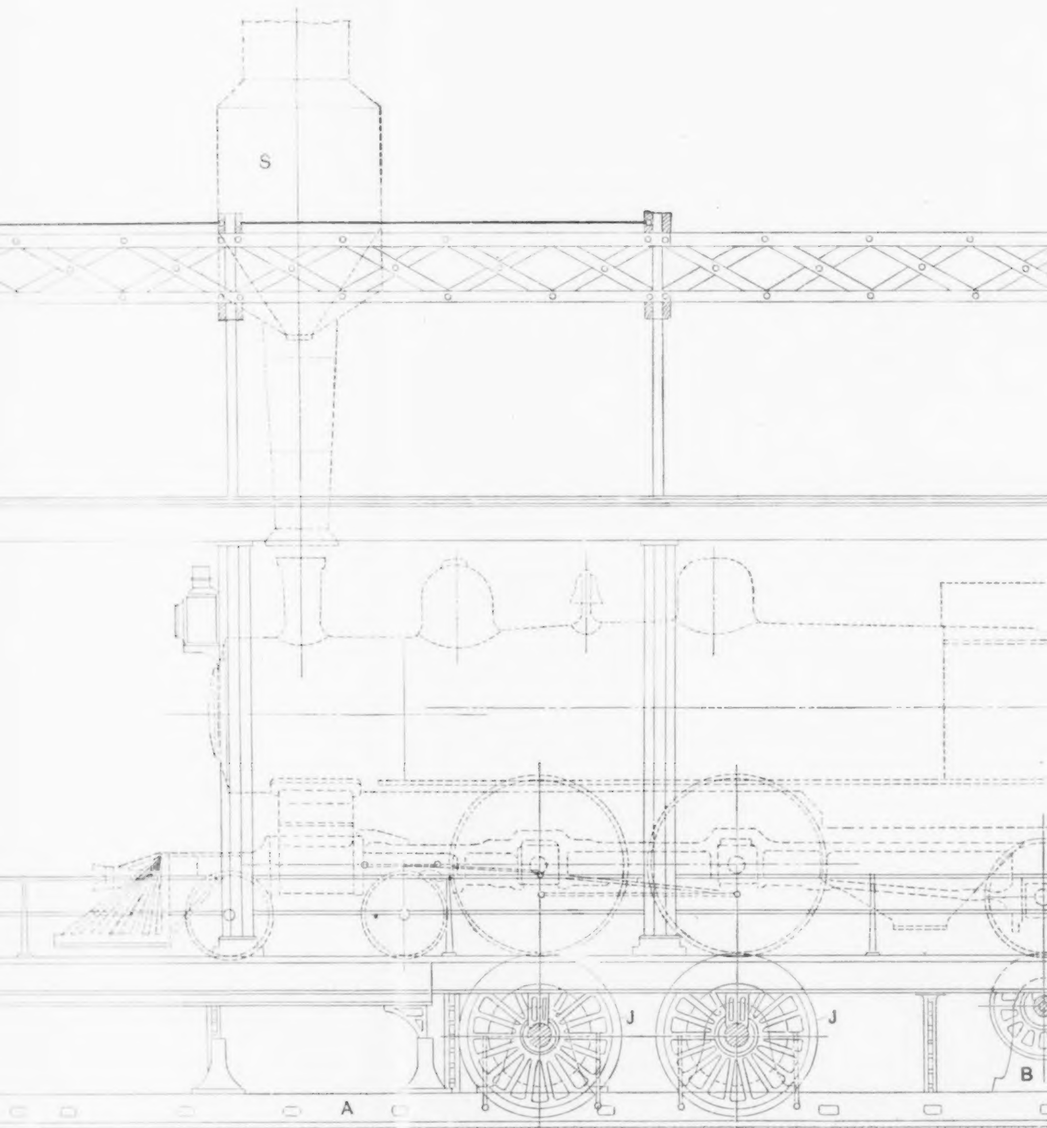


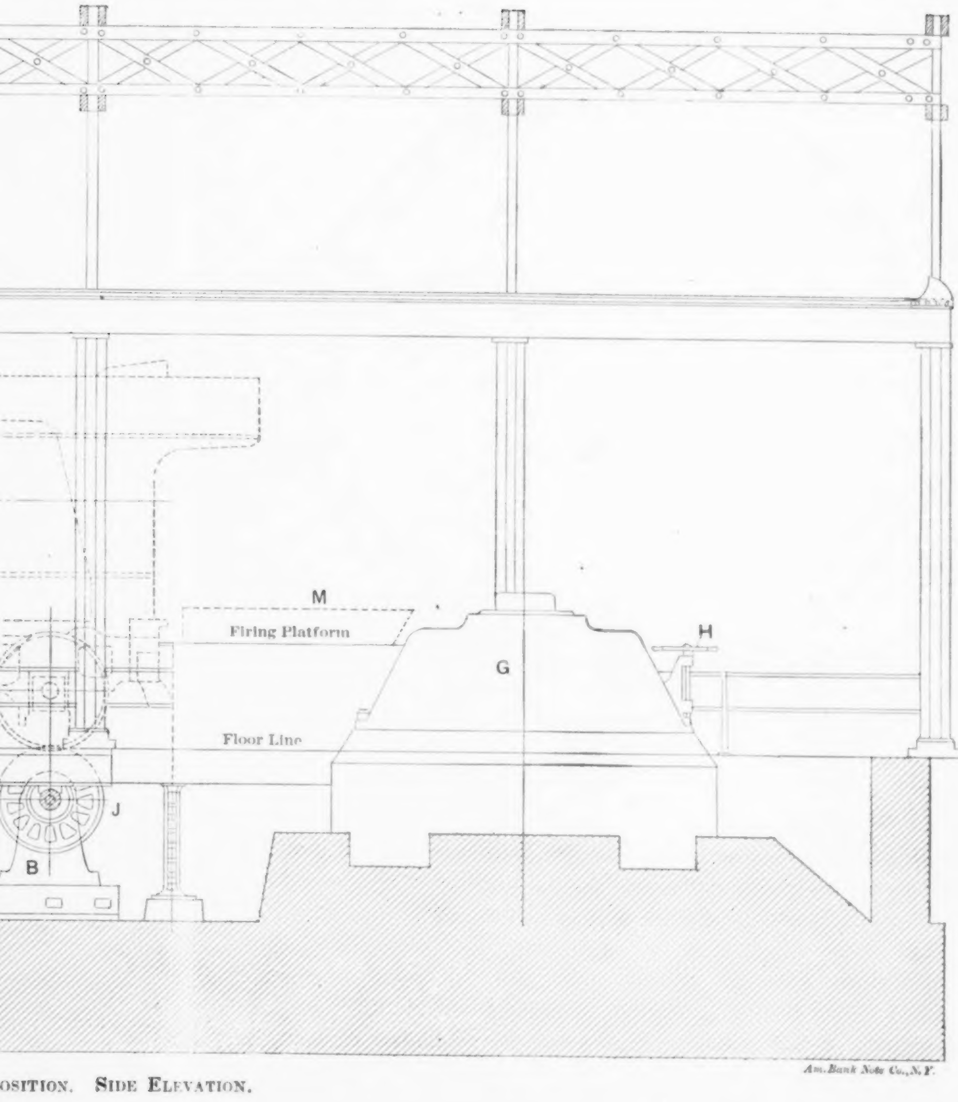
FIG. 424.—LOCOM



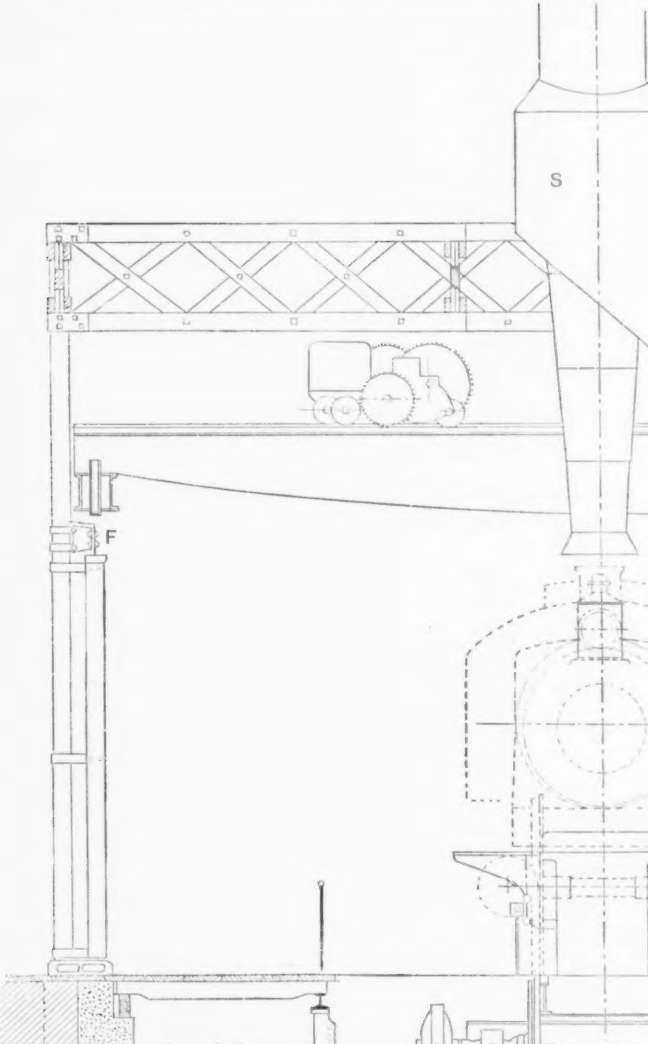
Locomotive Testing Plant of the Pennsylvania Railroad System at the Louisiana Purchase Exposition. Side Elev



Locomotive Testing Plant of the Pennsylvania Railroad System at the Louisiana Purchase Expositio

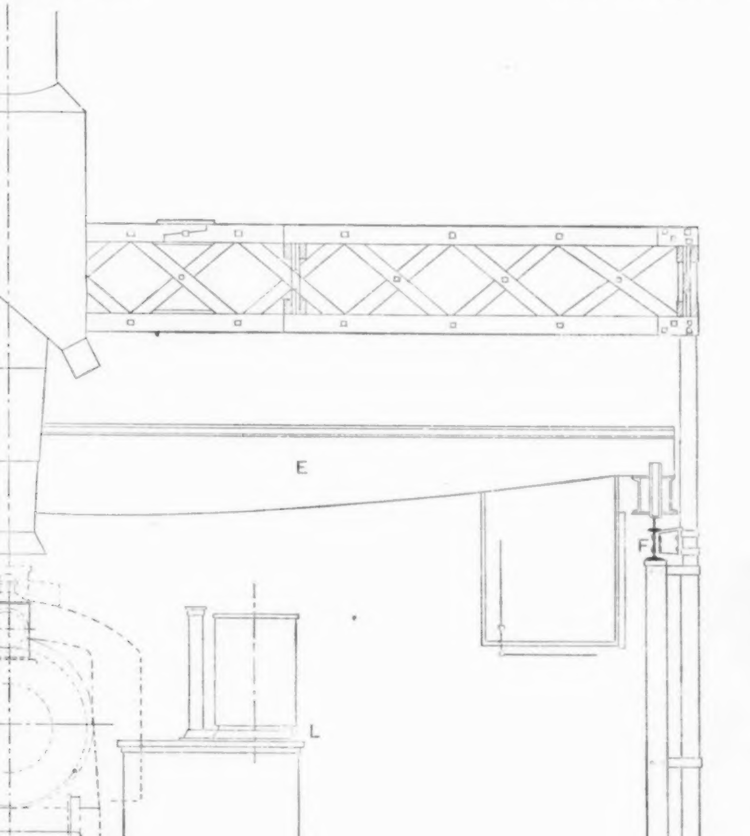


OSITION. SIDE ELEVATION.



Goss, W.F.M.

Fig. 425.—Locomotive Testing Plant of the Pennsylvania Railr



RAILROAD SYSTEM AT THE LOUISIANA PURCHASE EXPOSITION. END ELEVATION.

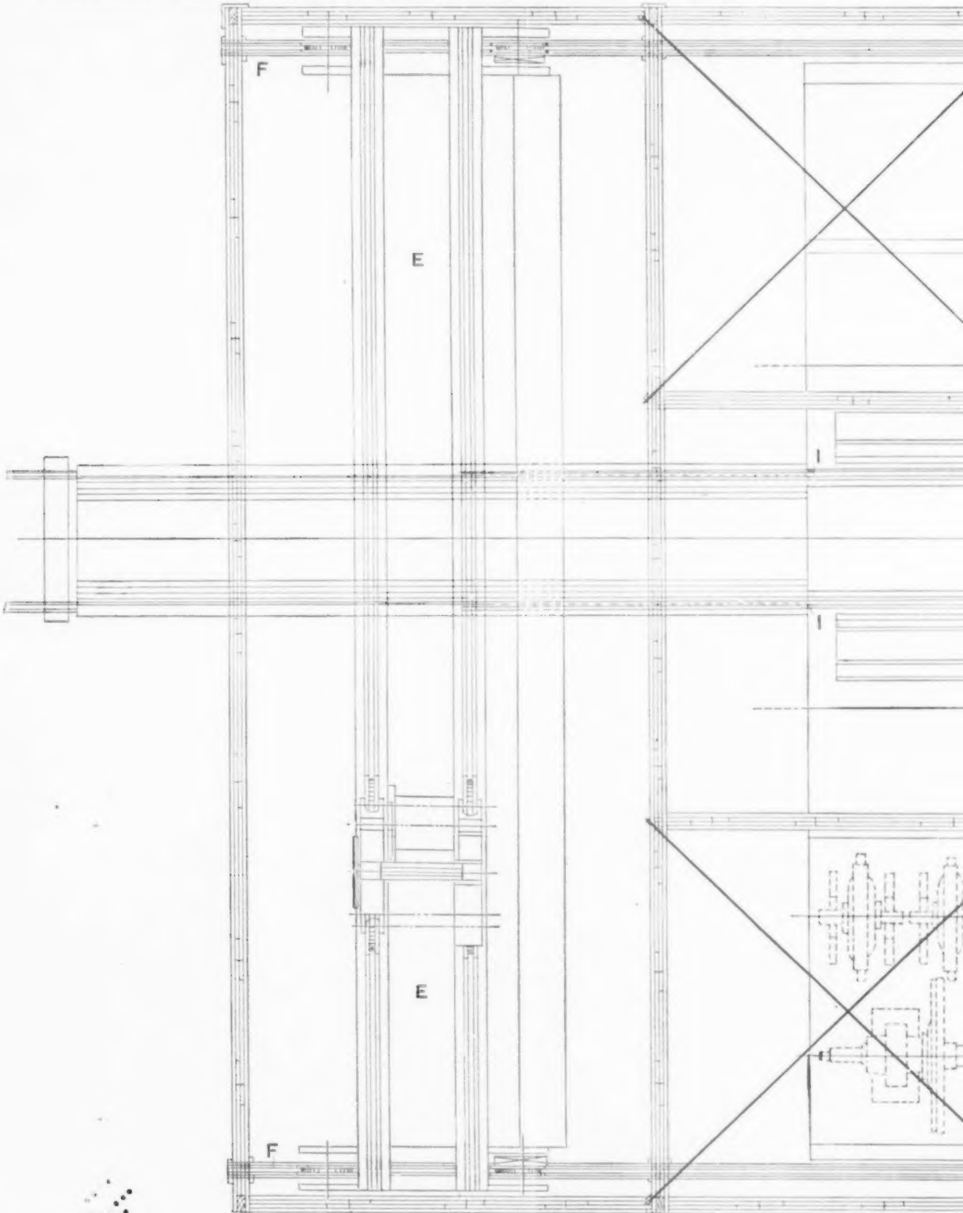
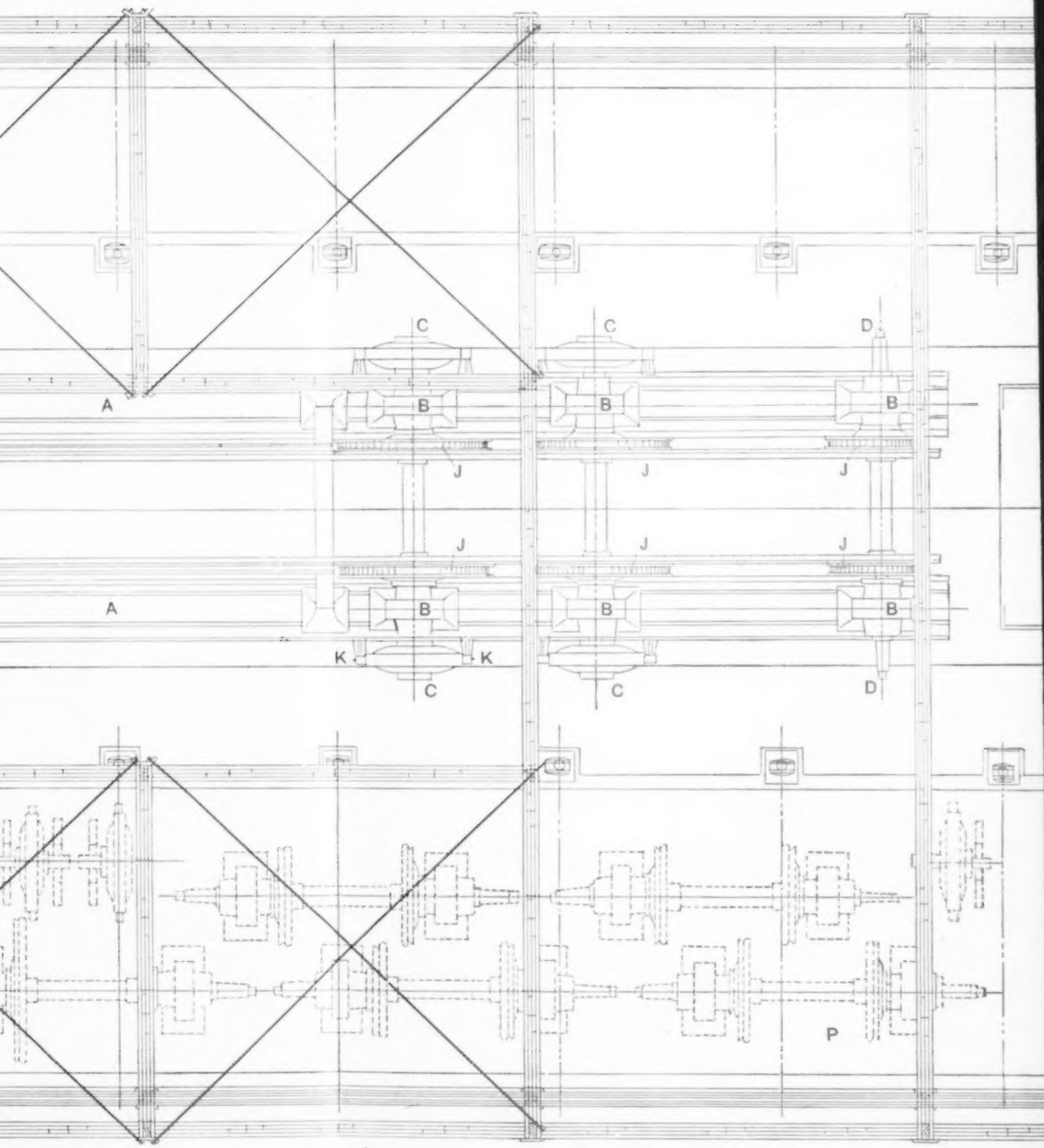
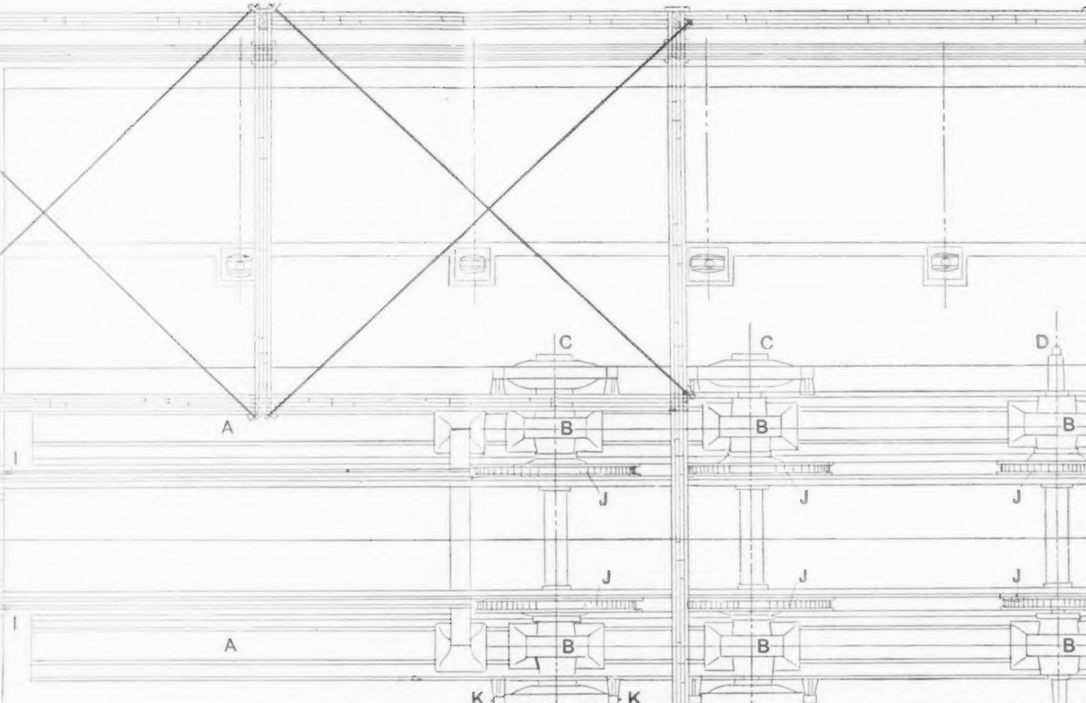


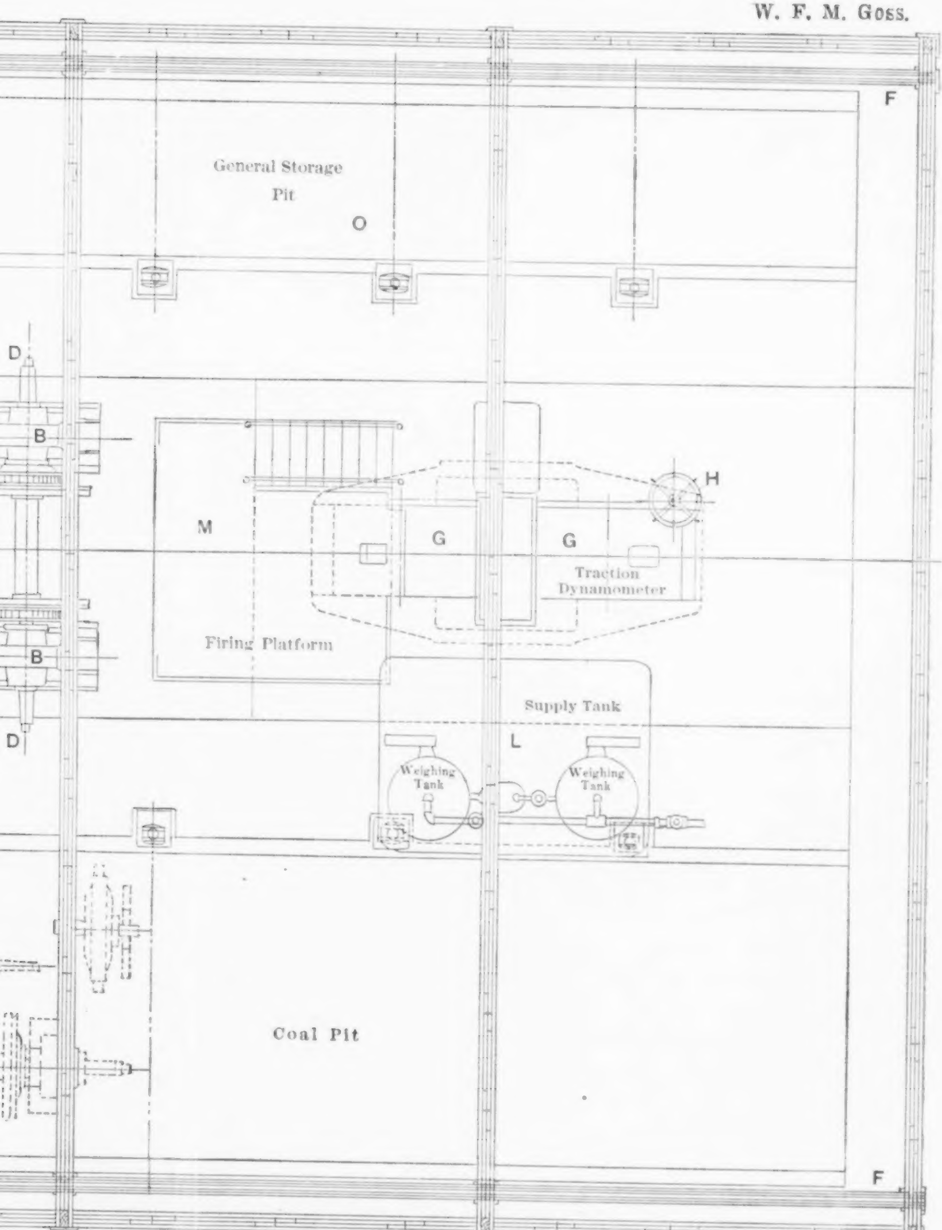
FIG. 426.—LOCOMOTIVE



OTIVE TESTING PLANT OF THE PENNSYLVANIA RAILROAD SYSTEM AT THE LOUISIANA PURCHASE EXPOSITION. PLA



6.—Locomotive Testing Plant of the Pennsylvania Railroad System at the Louisiana Purchase Ex



... Exposition. Plan.

Am. Bank Note Co., N.Y.

are segments of a circle having their centers at the center of the tube. The angular motion at the end of one arm imparts straight line motion to a carriage, guided by a grooved track and carrying the recording pen. The opposite arm is coupled by steel belts to a rotary oil dash pot, to reduce violent oscillations of the recording pen, the extent of which can be controlled as desired. The principal resistances in the dynamometer are flat springs, which may be changed to make the eight-inch travel of the recording pen correspond to a draw-bar pull of either 80,000 pounds, 40,000 pounds or 16,000 pounds as may be desired. The draw-bar pull and a datum line are traced upon a strip of paper 18 inches wide, made to travel at a known rate for each mile run by the locomotive. In addition to the pens tracing the draw-bar pull and the datum lines, five other pens are provided so as to indicate on the diagram each square inch of area as recorded by an integrator attachment; time intervals of one second; each thousand feet traveled by the locomotive; time at which indicator cards are taken; and one extra pen for any special record that may be desired. This diagram will form the permanent record of the draw-bar pull, together with the other information recorded upon it. The yoke of the dynomometer can be adjusted vertically through a range of 12 inches by means of a wheel *H* in order to accommodate different heights of draw-bars.

The smoke from the locomotive is carried out of the building by a stack, which can be moved longitudinally of the plant to any position required, and the lower portion of which is made telescopic, so that it can be raised and lowered for adjustment, and to permit the passage of the electric crane. The stack has deflectors, so that sparks discharged by the locomotive can be caught and weighed, giving a record which will form a part of the data obtained.

To keep constant the speed of a locomotive on the plant, there is a by-pass around the main valve controlling the supply of water for all the brakes, in which is an automatic valve, having the form of a throttling governor driven from one of the supporting axles. If the speed of the locomotive increases beyond the desired number of revolutions per minute, the by-pass valve opens, augmenting the water pressure on the brakes, and if, on the other hand, the speed of the locomotive falls below that desired, the automatic valve closes, and decreases the pressure on the brakes.

Facilities for securing observed data are unusually complete.

Coal is delivered to the locomotive in weighed boxes, within reach of the travelling crane, by which it will be handled one box at a time. Feed-water is measured in two calibrated tanks, mounted on scales to facilitate checking. These are filled alternately, and discharge into a third tank, from which the injectors take their supply. All water delivered to tanks is metered to supply an additional check. The equipment of such minor accessory apparatus as indicators, pressure recording gauges, pyrometers, calorimeters, and tachometers, is complete and of a high character. An adjustable indicator rigging is of such form as to permit it to be applied to any locomotive, and requiring only special designs of brackets suited to each individual case. A valve-diagram apparatus may be used at speed as well as when running slowly to record valve elipses, showing the extent and time of the movement of the valve.

It is the purpose of the Pennsylvania System to test during the period of the Fair twelve different engines, running from sixteen to twenty different tests upon each. Each locomotive tested will be representative of some particular general type. The significance of the undertaking may be judged from the fact that the operation of the plant involves a permanent staff of twenty-seven men, some of whom are concerned with securing data, others with working it up, and still others with the preparation of matter for publication. A complete outline of the work as originally planned by the advisory committee has been published as Bulletin No. 1 * by the Pennsylvania Railway System. This outline is now in the process of being executed. Bulletin No. 2 † describes the mechanism of the plant. Other bulletins describing the progress of the work will from time to time be issued, and at the conclusion of the test all will be combined to make up a single publication.

It is well to make of record the fact, that in installing its plant at St. Louis, and in the work of testing, which is now in progress, all involving an enormous expenditure of money, a single railway company makes a generous and most extraordinary contribution to science. In their appreciation of this undertaking

---

* Locomotive Testing Plant at the Louisiana Purchase Exposition. Bulletin No. 1: Organization, Plan and Scope. The Pennsylvania Railroad Company. The Pennsylvania Lines West of Pittsburg. F. D. Casanave, Special Agent, P. R. R., Philadelphia, Pa.

† Bulletin No. 2: Location and Description of Plant.

and that their work might have a wider direction than it could otherwise have, the officials of the railroad company early called this Society and the American Railway Master Mechanics' Association to its aid, asking that committees be appointed to assist them in their work. With the coöperation of these organizations, an advisory committee was formed, which has since served as a board of consultation to the railway company.* The Society can not fail to find satisfaction in having been given a part in so important a work.

In conclusion, the writer is pleased to acknowledge the important aid which has been rendered in the preparation of this paper by Mr. H. F. Wardwell.

DISCUSSION.

Mr. Edgar Worthington.†—Having spent the greater part of my life in designing and building locomotives, although I am not so engaged now, I feel strongly how much we are indebted

---

* The organization under which the work is being advanced as set forth in Bulletin No. 1 is given below. It is expected that the advisory committee will be increased from time to time to include certain official representatives of foreign governments.

The Pennsylvania Railroad System:

J. J. Turner, Third Vice-President, Pennsylvania lines west of Pittsburg; Theo. N. Ely, Chief of Motive Power, Pennsylvania Railroad System; F. D. Casanave, Special Agent, Pennsylvania Railroad System; E. D. Nelson, Engineer of Tests, Pennsylvania Railroad Company, Altoona, Pa.

Louisiana Purchase Exposition:

Willard A. Smith, Chief of the Department of Transportation Exhibits, Louisiana Purchase Exposition.

ADVISORY COMMITTEE.

On Behalf of the American Society of Mechanical Engineers:

W. F. M. Goss, Dean of the Schools of Engineering, Purdue University; Edwin M. Herr, General Manager, Westinghouse Air Brake Co.; J. E. Sague, Mechanical Engineer, American Locomotive Co.

On Behalf of the American Railway Master Mechanics' Association:

F. H. Clark, Superintendent of Motive Power, Chicago, Burlington and Quincy Railway; C. H. Quereau, Superintendent of Shops, New York Central and Hudson River Railroad; H. H. Vaughn, Assistant Superintendent of Motive Power, Lake Shore and Michigan Southern Railway.

OFFICERS.

F. D. Casanave, Special Agent of the Pennsylvania Railroad System; W. F. M. Goss, Chairman of the Advisory Committee; H. H. Vaughn, Secretary of the Advisory Committee.

† Member of the Institution of Mechanical Engineers.

to Professor Goss for the work he has done in the last ten years and should like to express how much locomotive engineers on the other side and in Europe generally appreciate the tests which he has made at Purdue University. Many a time have I gone to the tests of Professor Goss, more especially the beautiful tests which he made of pressure on the rail. Several locomotive superintendents in England expressed to me their regret that they were not able to be present, but unfortunately at this time of the year in England they are very busy, and I can assure Professor Goss that they agree with me that we owe him a very great deal, and you may judge the genuineness of their admiration by the way Mr. Churchward is imitating his example.

*Prof. W. F. M. Goss.**—Since the foregoing paper was presented and discussed I have made the acquaintance of Mr. Michael V. Gololoboff, who has been engaged in the development of a locomotive testing plant at the Putiloff Works, in St. Petersburg, Russia. I am sure the details of this plant would interest the members of the Society, and, by the courtesy of Mr. Gololoboff and his assistant, Mr. S. T. Smirnoff, Director of the works, the following description is abstracted from a larger presentation.

The plant is supposed to be the first of its kind in Europe. Its plans were started in June, 1903, and the plant was practically completed May 1, 1904. It occupies a space in the erecting shop.

The plant proper provides six pairs of supporting wheels, all of which are provided with Alden friction brakes, and it is of sufficient size to accommodate the largest locomotives, not excluding the Mallet articulated compound of the B. & O. The supporting wheels are 49½ inches in diameter, and can be adjusted by means of slots in the bed plates to accommodate engines with a maximum driving wheel base of 27 feet 11½ inches. The supporting pedestal can be moved by two workmen with bars working into racks formed in the bed plates. This arrangement is similar to that employed in the testing plant at the St. Louis Exposition and designed by the Pennsylvania Railroad.

The water for the brakes is supplied from a steam pump through an accumulator. Differences in pressure can be secured by varying the quantity of water in a tank on top of the plunger and by using counter weights. The valve in the steam pipe of the pump is controlled by the movement of the plunger.

* **Added since the meeting.**

A governor by regulating the water pressure in the brakes can be set to maintain a standard speed. A section of rigid track is provided between the supporting wheels. It is made of built-up I-sections, the rails having grooves in which the flanges of the locomotive run. The track is supported at the middle by screw jacks and at either end by tapered blocks; these blocks can be moved inward or outward, either raising the track into position or lowering it out of the way.

The traction dynamometer is in two independent parts, which may be called the receiving head and the measuring apparatus. The former is essentially the same as that employed at the Purdue testing plant. The latter was designed by Mr. Gololoboff to keep a complete record of the tractive force exerted by the locomotive during the test. It consists essentially of a small oil sack similar to that employed in the receiving head, against which a piston acts. The pressure on the sack acts through the piston through levers steadied by a dash-pot and a counter load, and its motion is transmitted to the indicator of the gauge. The latter is so scaled that one division of the arc equals 10 kilos., and from it the tractive force can be read at any time. From the dial motion a pencil recorder gives the intensity of the pull. The amount of water used is measured in two graduated tanks. A third tank is employed for the measurement of oil, when this is used as fuel.

The coal is stored in a large hopper outside the building and enters into a weighing bucket through a chute. The smokestack is made in two parts, so that a vertical adjustment can be secured.

The accumulator is one of the most salient elements of difference which this plant offers from others of similar sort. It is further provided with all accustomary apparatus for complete tests, and if satisfactory results are obtained these tests may replace those on the road. The Russian state railroads and the Southwestern Railroad (Vieff) are considering the erection of similar plants, but as yet no work has been started on either.

No. 1040.*

A RATIONAL BASIS FOR WAGES.†

BY HARRINGTON EMERSON, NEW YORK, N. Y.
(Member of the Society.)

1. Both the coal operator and the wage earner know better what they are selling than the average manufacturer knows what he is buying, whether coal or labor, and as a consequence the latter usually has the worst of the bargain, which is wholly his own fault. I couple coal and labor because there is much ultimate similarity between them as objects of contract.

2. The coal operator aims to sell tonnage. He owns a mine, whose quality he cannot change; he mines by the ton; he pays freight by the ton, and if you will let him, he will sell by the ton, persuading the railroad, factory purchasing agent or other customer that coal is coal; no difference except in his price, which he is willing to shade a little below that of competitors.

3. The purchaser who knows his business is not deluded by a ton price, particularly not if he is running a torpedo boat or cruiser on a premium trial trip. He buys what scales do not weigh, namely *power*, and if the factory manager can get

---

* Presented at the Chicago meeting, May and June, 1904, of the American Society of Mechanical Engineers, and forming part of Volume XXV of the *Transactions*.

† For further discussion on this topic consult *Transactions* as follows:

No. 341, vol. x., p. 600: "Gain Sharing." H. R. Tonne.

No. 449, vol. xii., p. 755: "Premium Plan." F. A. Halsey.

No. 647, vol. xvi., p. 856: "Piece Rate System." F. W. Taylor.

No. 909, vol. xvii., p. 1040: "Drawing Room and Shop System." F. O. Ball.

No. 928, vol. xviii., p. 341: "Bonus System for Rewarding Labor." H. L. Gantt.

No. 965, vol. xxiv., p. 250: "Gift Proposition for Paying Workmen." Frank Richards.

No. 1001, vol. xxiv., p. 1302: "Machine Shop Problem." Chas. Day.

No. 1002, vol. xxiv., p. 1322: "Graphical Daily Balance in Manufacture." H. L. Gantt.

No. 1903, vol. xxiv., p. 1337: "Shop Management." Fred. W. Taylor.

power by the metered kilowatt he will forget all about coal. If he cannot obtain measured electric power or water power, he must create his power by means of a heat engine, and here again he is lucky if his fuel can be metered to him in cubic feet of natural gas, or in barrels of crude oil, both of these substances having a very accurate number of heat units, for the gas, per cubic foot at constant temperature and pressure; for the oil, per gallon or pound.

4. In comparing the relative value of coal, oil or natural gas, the consumer can only be guided by the cost in dollars of brake horse-power, and if he is wise he will purchase coal, by the ton, perhaps, but strictly on a basis of dollars per thermal units.

5. When a ton quotation is made, the sensible purchaser adds to the delivered price the cost of unloading, shrinkage, cost of delivering to furnace room, cost of removing ashes, thus obtaining the total cost of the coal as used in the furnace. He will then analyze the coal, make a calorimetric or experimental test of its heating value and thus determine the relative cost of the heat units in the coals offered. If he does this he will invariably find that the best coals are the cheapest. It costs on the average no more to mine poor coal than to mine good coal, probably not as much; it costs no more to carry to market good coal, and the man operating a seam of good quality is just as anxious to increase his tonnage, and thus add to his margin of profit, as the owner of a poor seam. As the majority of dealers and purchasers make but little allowance for difference in heating value, the owner of the best seams has to shade his price almost or quite to the level of the price of the poorer coals of his competitor for *tonnage*.

6. Within the last year I have seen a large concern buy at over $6 a ton, imported English coal of 11,000 British thermal units—coal with which steam could not be maintained, and later in the same year this firm made chemical analysis and calorimeter tests of coals offered, securing for $3.10 a coal of 15,300 British thermal units. The first coal was bought because at the time it seemed cheap per ton; the second was bought solely because it was low in price per heat unit. It is evident that two coals do not always have the same relative value at different places. At Nome, in Alaska, I have seen coals sell at $75 per ton, owing to the cost of transportation, the expenses of landing, of carrying inland, of dealers' profits. If, in a camp near Nome,

a coal of 10,000 British thermal units is worth \$75 a ton, then a coal of 15,000 British thermal units is worth \$112.50 a ton. Deduct from both the cost of delivery from Seattle to mine, say \$71 a ton, and it becomes evident that the purchaser should not have hesitated to pay if necessary \$41.50 for the better coal rather than \$4 for the poorer fuel. I knew but one man at Nome far-sighted enough to realize that what he was paying for was heat units, so he imported and used crude oil.

7. At the furnace door the connection between dealer and buyer ceases. After having sold British thermal units, the dealer is not further concerned as to what use the buyer makes of the coal. One buyer will put in the best boilers he can secure, install a condensing, triple expansion engine and obtain a horse-power from one pound of coal. Another buyer will with the same coal be content to obtain a horse-power for seven or eight or even more pounds of coal.

8. In the economical consumption of coal *three* considerations enter:

(1) The market price per ton;
(2) The quality of the coal;
(3) The efficiency of use;

and all three must be considered to obtain heat units cheaply.

Prices are quoted per ton to the purchaser, who should determine qualities and buy on a basis of heat value and not of weight, and finally he should use what he has bought so as to obtain highest results.

9. What would we think of the dealer who demanded a higher price per ton because the purchaser had better boilers and engines? Yet, on the other hand, the dealer is debarred from asking a price based on his heat units because his mine conditions and competition force him to sell by the ton. The buyer, not the seller, is in the stronger position.

10. In contracting for labor the manager should consider it just as he does coal and buy it in the same rational and unusual manner.

11. The wage earner, like the coal operator, is forced to sell what he always has; namely, time. The employer, who is not interested in fuel tons, is also not interested in the wage earners' hours and minutes, but solely in low cost of production, and this he can only secure through the quality and efficiency of the labor he employs, only indirectly and remotely from the hours

the employe gives him. The wage earner is not selling *output*, nor is the employer buying *output*, and any attempt to base wages on *output* is as irrational as to base the price of coal on boiler and engine efficiency.

12. There was a time when labor was in a condition of status. The worker was entitled to a living more or less generous, and under pressure he turned out more or less work, just as sheep pasture their owners' ranges and peacefully submit to a shearing. By political economists and students of sociology the substitution of contract for status has been heralded as a great advance, just as the substitution of coal bought by the ton for wood gathered without measure from the forest was an advance. Under contract the wage earner is supposed to receive an agreed upon wage and to render a definite service, just as the ton of coal is supposed by those who know no better to be a definite measure of heat supply. This crude basis of demanding a surrender of time and leaving the particular service to the whim of the employer worked fairly well with certain passive services, in which time rather than effort and intelligence constituted the equivalent. Time reward was probably originally based on military or naval service, in which long periods of idleness alternated with hours or days of frenzied effort, much as firemen are employed to-day—a profession in which pay cannot be gauged by performance. It is also probable that many of the earlier wage earners were employed more for the sake of ostentatious display than for what they produced, survival of which we still see in the barbaric splendor of two postillions, a coachman and two footmen, all in gay livery, taking one poor man or woman driving. It is time the industrial world moved away from such prototypes, whether military or ostentatious, and by advocating an entire departure from the military type of management for factories our honored member, Mr. Fred W. Taylor, plans a reform far reaching and deep.

13. Military and naval idleness, ostentatious time-serving, are not wanted in modern intensive production and anything of the kind should be more obnoxious to the self-respecting producer, who is directly debased by it, than to the employer who cannot help but feel in a measure exalted by considering himself a captain of hundreds or of thousands. If a non-progressive father bequeathed to a progressive son an establishment with 1,000 employes, and the latter was able by the introduction of better machinery and modern methods to turn out more and better work

with half the men, even though he doubled their wages, he might debase himself socially.

14. There is a mental laziness in talking about dollars and days as if either were a humanly measurable quantity. The cost of living may go up or down, and I have heard men say that there were certain five minutes in their lives which they would not exchange for other months. When the wage earner sells time the employer does not necessarily obtain anything, not even as tangible an asset at a ton of very poor coal. Just as the coal operator has foisted tonnage on his client, so the foolish employing class has forced the wage earner to foist *time* on his employer, time of no value in itself, and the employer blind to the essentials of the transaction builds up an elaborate system of time checks and time keeping (both of which are merely incidentals and not essentials in good system) to the utter neglect of matters much more important.

15. Some thinking employers, realizing that *time* in itself had no value, have endeavored by means of piece work to buy *output*. The analogy of coal has already shown that *output* does not enter properly into the contract between employer and wage earner. When he sells his time, the wage earner does give something of value to himself, even if it may have none for his employer, but he can no more accurately and justly sell output than the coal operator can sell horse-power from a boiler and engine over which he has no control. It might on occasion be convenient temporarily to measure the merit of the wage earner by his output, just as it might be convenient to measure the value of a coal by the horse-power obtained from it in a given power-house, but the moment a new power-house is put up, the moment a wage earner is coupled up with better facilities, the purchaser falls back on the market price of coal and on the market price of skilled labor.

16. In buying coal the wise purchaser was advised to buy by the ton, to base his price on quality and to use efficiently. He should act similarly when he buys labor.

17. It is perfectly possible to find an equitable basis for reward, at once just to the employer who wants something more tangible than the abstraction *time*, and to the wage earner who is already too precariously dependent on uncertain wages, when what he really wants is a steady living. The wage earner, like the lawyer, has four different things to sell:

(1) His time and liberty.

(2) His skill, profession or trade.
(3) His intelligent co-operation.
(4) His power to do harm.

18. *Output* is not among these nor is it, except in rare, and general distressing, cases, like sweat shop work, what he does or can sell.

A living daily wage is due to every man who signs away his liberty to another, whether he stands around idly to glorify his employer or to amuse himself.

19. This minimum wage is a matter of contract. It should be due the man whether he is present or absent as long as he remains on the pay roll. This is the wage payable as equivalent for assignment of time and loss of liberty. This part of compensation I shall call minimum wage. Above this minimum he should receive when at work at his trade a different and larger sum, equivalent to the current wages of his trade but in reality made up of two distinct parts, namely, the minimum wage and an increment due to him for his profession or trade. We have in this the basis of the present daily wage system relative to time. Beyond this, however, for future just and peaceable relations between employer and employee, comes the most important part of the wage earner's reward, namely, what is due him for exceptional and unusual co-operation either of mind, body, or both. I believe that in the old days of privateering the sailor who first sighted a vessel subsequently captured was entitled to special prize money, and in war, actions of exceptional personal gallantry are specially rewarded.

20. From the nature of the case this third increment of wages, corresponding to the contingent fee of the lawyer, must be fluctuating, payable some days and others not, because earned some days and others not. In many cases the earning of this professional wage does not imply harder or most exhausting work, but more intelligent and coöperating work. It implies use of heart and head in addition to use of hands and eyes. We might say that these three forms of wages are paid respectively:

(1) For use of body.
(2) For use of hands and eyes.
(3) For use of head and heart.

21. It is foreign to this paper to discuss the fourth form of wages due to the ability to do harm. This wage may be no better than the extortion of the blackmailer or the ransom of

the brigand, or it may be as legitimate as the fee paid a lawyer to keep out of a case.

22. As the State fixes the minimum age and the maximum hours at which children may be employed, it might also, at least by example, fix the minimum living wage below which no one should accept employment, and thus fix a minimum standard. Supply and demand, modified by some extent by the Unions, will determine the second wage, and the third is a matter to be left wholly to employer and employee, even as are to-day lawyers' and doctors' fees.

23. The employe has everything to gain by a system of strict and accurate records which will enable him to establish his own value and worth, and on the basis of which, if he changes location, he may be eagerly sought elsewhere. Whoever heard of a lawyer, when he moves to a fresh location, taking with him from the mayor of his old town a certificate that he has left honorably? The official court records show whether he has won cases or lost them; whether he has been a blackmailer or faithful to his clients.

Why should not the machinist going to a new location say: "Here is the record of work I have done. I know so well how to do what is wanted; to combine machine performance with best cuts, feeds, speeds, as to lower materially the average cost of production of any operation I undertake. I expect, firstly, current machinists' wages, and, secondly, I expect so much more for my special skill."

24. The theory and application of triple wage can best be illustrated by a concrete example, for which I shall select the case of a locomotive engineer—a man belonging to one of the strongest and most conservative unions, receiving high wages owing to long training, special skill and great responsibility.

25. Such a man should be allotted a fixed daily rate of minimum wages, to be paid whether he runs or not, whether he is sick or well, whether he is suspended or in good standing. The amount of this daily wage is determined by agreement at time of receiving appointment.

26. Secondly, under the usual terms of his own or his union's agreement with the company, he receives when on his runs definite and agreed upon compensation due to him because he is an engineer in charge of a train and under orders, whether standing on side track or running sixty miles an hour. There is, however,

a great difference in engineers and in firemen, who should be jointly considered with engineers. One engineer and his fireman will send into the shop in short time every engine they run on. Another engineer and fireman succeed in getting the utmost out of their engine even under the most trying conditions of service. It is just here that the difference comes in between man and man for which at present there is scant, or no recognition, a difference that corresponds to difference in heat units of coals.

27. In one of the largest of American railways the cost per mile for wages of engineers, firemen and wipers, the cost per mile for coal and the cost per mile for locomotive repairs is practically the same, being respectively \$.0946, \$.0999 and \$.0867. This is an average for the whole road, but for the heaviest engines both repairs and fuel are relatively greater. It is in the second and third items that there is abundant opportunity for railroad companies and their engine crews to combine to effect a saving in which both are to share, and the relative rights of each party are not to be guessed at or conceded as a gratuity or bonus, but are to be based on scientific detail study as reliable as the determination of heat units in coal or the efficiency of a boiler or engine. What encouragement is there to an engineer to work and toil over an engine that some other engineer has shamefully neglected? What encouragement is there to him to try to put an engine into the best of condition if the next man that takes it will allow everything to run down and go to ruin?

28. An engine costs twice as much to-day as a few years ago, and four times as much is expected of it. The wear and strain are in every way greater, and the repairs have become more costly. It is easier by omission of *unusual* care to cause damage, more expensive to make it good, yet these new conditions have not been met by any corresponding change in the relation between employer and employe. I leave to others who know more about this particular subject of engines than I do to evolve an equitable basis of mutual gain to both parties. I can imagine, however, that it might be possible to give the man with the best record the pick of certain engines, the selection of his own fireman, and in addition to paying him by the day and by the regular run basis, to give him an ever-increasing rate in proportion to the number of miles his engine runs without getting into the shop. When at last the engine has to be sent there he is laid off on the day wage basis, possibly with the privilege of superintending

the repairs, until it goes into service again. I can think of other methods of accomplishing the same desirable end; namely, securing the earnest, constant coöperation of those in charge of the engine, in keeping it in good repair.

29. Companies know the average run of their engines, the average cost or repairs per mile run, the average earnings of the engineers and firemen. All the data are at hand to put into effect the three rates of wages.

30. It will at once be objected that engines must be pooled in order to make more miles than they will if assigned to one crew. This objection may be urged against this particular example, but not against the principle urged, yet even for the example no one who has not studied the subject thoroughly and exhaustively has authority to speak with finality. I know of no data available. It is a curious fact in railroad and other machine shops that clerks and facilities are lavished for *money* accounting but elementary help begrudged for shop records; also that betterments and improvements that would yield 100 per cent. profit on their cost are not made simply because the records *pro* and *con* do not exist. It would be possible to work out this engine crew problem so as to obtain maximum of engine service, maximum care of engines, maximum wages to crews, minimum of repair costs and lessened expense to companies. On large ocean steamers engine crews alternate, not promiscuously but definitely, and on a railroad there need be no hard and fast rule one way or the other, but a scientific adjustment of equipment to conditions, so that some engines would have only one crew, other two engines have three crews, with chief engineer, and first and second assistants, and some few emergency engines have no regular crews. These matters require trouble and study, but so does the running of trains on time. The latter part of railroad operation has reached a high state of perfection, but the adjustment of engineers, engines and company interests to one another is as yet very elementary.

31. A system that is applicable to engineer and fireman is also generally applicable to other craftsmen. It is a principle I advocate, not an isolated case, each one of which will vary in method.

32. There are only two ways of obtaining the lowest cost of production: first, a relentless and ceaseless study and direction of details by the officials who plan and control, and, secondly, an

intelligent and eager coöperation of the employee, which cannot be equitably asked for without generous special compensation. It is simply a question of what is most profitable.

33. In industrial matters, whether production or wages, there is no possibility of permanent stagnation. Whether we export or import, whether we depend on home markets or on colonies, prosperity is ultimately based on the world's crops, and these depend on the variable rains from heaven. There are fat years and lean years, and it is only such a one as Joseph who is able to use the fat years to corner the grain market and the lean years to enslave the Egyptians forever. When work is abundant, employment increases, wages rise; when work is slack, employment decreases and wages fall, a double loss to the wage earner—dearth of employment being far more serious than lower wages, and in this respect the census and other reports dealing with rise and fall of wages have been most misleading. The question is only very slightly whether the average wages per year of those actually employed are higher or lower, but whether more or less are employed *in their own trades* at reasonable wages.

34. It is a reflection of the intelligence of the State that when so much is done to correct nature, to dredge harbors, build sea walls, impound waters, regulate rivers, fight against diseases; when so much more is meddled with that might be better left alone, absolutely no use is made of the great power of the State to act as a fly wheel for the energy of production. Here at least is something in which the interests of employers and employees are one. When labor is scarce and materials high, governments, national, State and municipal, should carefully abstain from undertaking great works of creation or improvement, but when labor is plentiful and materials low in price governments should carry out plans held in reserve for just such conditions. There is no other means at once so powerful and economical to minimize the ups and downs of both labor and capital. It is in the power of government to establish a minimum wage (as well as a minimum price for great staples) at which it is always ready to undertake great elementary works of public utility—dredging canals, for instance, or opening roads. In some such manner as this would I have the State take a hand in helping to fix the minimum wage.

35. On the other hand every wage earner should keep constantly in view the possibility of obtaining a much higher rate

than the average; a rate wholly due to his own reputation and accomplishment; a rate far above the minimum offered by government, also much above the mean due him as an able-bodied, skilled, trustworthy craftsman. This extra rate must always be based on the fact that he is more skilful or valuable than the average. He may be more skilful to-day, just as a prize fighter may be the champion of the world to-day, but next year both may be no more than mediocre. He must constantly excel if he would constantly command a professional price.

36. It is, however, not necessary to wait for the slow action of an unintelligent and unmindful government. Every employer can test the cost of production as he can the cost of horse-power, and he should make it his business to know what he can afford to pay for quality; and the employer who systematically seeks quality will find that the best coal and the most efficient labor sell for less than their relative worth.

37. Believing as I do that the man, machine and method efficiency of the average railroad or machine shop is far below what it might be, often not more than one-third to one-half of the ideal, I cannot but regard disputes over wages as the effect rather than the cause of unsatisfactory and unscientific relations between employer and employee. If the employers had been wiser there might perhaps never have been any union antagonism, only reasonable bargaining as there is to-day between lawyer or doctor and client. It is the employe's privilege to try to obtain higher wages and shorter hours, but not as an inducement to offer *less work*, nor yet, perhaps, more work, but *better work*, a more efficient combination of man, method and machine.

DISCUSSION.

*Mr. Emerson Bainbridge.**—I think I am right in saying that as in all engineering work, the question of wages enters into cost to the extent of forty to eighty per cent., it is very proper at a meeting of this kind to discuss a paper which bears upon this important subject. But the paper itself strikes one as containing many very extraordinary suggestions. The paper might be described as an irrational argument tending very much to confuse the question of the difference between the position of the capitalist and his work-

* Member of the Institution of Mechanical Engineers.

ingmen. Indeed, throughout the whole paper I believe I am right in saying that it assumes that the employer is an uncommercial, inexperienced, foolish person, and that the workingman is a downtrodden, ill-used, ill-paid servant. As a large employer of labor I must say that I find myself entirely unable to agree with hardly any sentiment expressed in the paper. It seems as though the writer had suddenly awakened to a condition of things which those of us who employ workmen largely have been alive to for a long time. For instance, he speaks of a purchaser not realizing the value of the coal or coke he purchases. Take the coke supplied to a blast furnace. Surely the first thing a buyer does is to make quite certain that he agrees with the producer what the percentage of sulphur, what the percentage of water, what the percentage of ash, and what the percentage of phosphorus should be, and he takes good care, having ascertained this, to base the price he pays accordingly. Now, this paper assumes that this common sense does not exist.

If this paper had been prepared with the object of suggesting some means of getting the best out of labor, I think it might have had a practical value.

Paragraph 36 refers to the slow action of an unintelligent and unmindful government.

Does this refer to the American Government or to the English Government? At first I thought it applied to our Government, because we have a Government that thought nothing of spending a million and a quarter a week for two years and a half in a war, but shrinks from spending a million upon education. I suggest that both here and in our own country nobody is better appreciated than the man who does his best for his employer and puts his best endeavor into his work. I am reminded of a mill owner in Lancashire whose mill suddenly stopped and a thousand men were left idle. He tried to start his engine, but could not, and he did not know what to do. So he sent for an expert. The expert came; he looked at the engine, and then called for a hammer, gave one stroke in the right place, and immediately the big wheel revolved and the mill again commenced work. Having got his men at work again, the mill owner told the expert to send in his bill. The bill was sent in and it amounted to five pounds and two shillings. The mill owner thought it too large, and he wrote for a bill of particulars, and he got it as follows:

"To mending your engine, 2s.

"For knowing how to do it, £5."

Finally, I suggest that the man who knows how to do it, both in this country and in ours, has his service well appreciated and well rewarded.

*Mr. E. J. Chambers.**—It is only a word or two that I have to say about this paper. I must say I was very much astonished to find that it came from a member of the American Society of Mechanical Engineers, because had it come from the old country, where we are supposed to be a bit slow, one wouldn't have been surprised. It reminds me of the feudal times. However, our friend has certainly given us something to think about. When I found the way in which he divided up the work of the laboring man, and the reasons that he gave, I was very much staggered. First of all, he speaks of the assignment of time and the loss of liberty. Well, gentlemen, I had no idea that we were coming back to the times of serfdom. Among our 1,100 or 1,200 men I do not think there is one of them who has assigned his liberty. What I think he really has done, and I believe most practical men will bear me out, is that he has joined me in a partnership, and therefore there is no question of liberty having been given up. When I notice how the workingmen look after themselves (I think very much better than their employers do), I believe the question of assigning their liberty to us is quite a misnomer. With regard to what the author says in respect to the way of calculating payment, there is nothing new about this arrangement. I remember when I was an apprentice, some forty-five years ago, one of our stock tales was that one of the men came and asked for a job and the Master said that he would give him sixteen shillings a week, and the man said, "Why, it takes fifteen shillings to keep me honest, and surely I ought to have more than a shilling for my work." Evidently that man forty-five years ago realized that there was a certain amount required to keep body and soul together and to keep him honest, and the rest he wanted for his work. Then we get another point in the paper, and that is that the workingman is not only to be paid when he renders his services, but he is to be paid that amount whether he is ill or well able to do his work. What a delightful arrangement, to transfer all the responsibilities

* Member of the Institution of Mechanical Engineers.

of keeping our work people in good health on to the shoulders of the already somewhat overburdened employers!

I do not think that is the way the business men are going to look at this question, and I do not think that this paper will meet with the slightest acceptance even from workingmen.

*Mr. J. Hartley Wicksteed.**—I wish to congratulate the author of this paper upon having made what seems to me to be a most excellent analogy between the use of coal and the use of labor. He says in the paper something to the effect that a user of coal does not pay, or ought not in justice to pay, according to the results that he gets out of that coal, because the result that he gets depends in very large degree upon the efficiency of his boiler and engine, and that similarly an employer ought not to gauge the value of the workman by the output of the factory, and "any attempt to base wages on output is as irrational as to base the price of coal on boiler and engine efficiency." I think that is a very shrewd observation and a very far-reaching point, that you may have a place so well managed that the general result is good, although the labor and the contribution made by each workingman is nothing more than usual, and it seems to me to touch the weak point of the scheme of profit-sharing by the workingmen, because it occurs to me that in any business with which I am acquainted the profits depend so very largely upon the skill with which the manufacturer makes his contracts, buys his material, and keeps up a steady flow of work through the shop, to say nothing of the difference between organized management that brings the work steadily and conveniently to the workingman's hands to turn out—all of which things are entirely beyond the immediate power of the workingman. I think that profit-sharing is a very rough way of tempting men into a shop by the prospect of sharing in prosperous times, and with the prospect of being highly discontented if they do not obtain a premium whether they earn it or not. Evidently the right way is to value the services of the workingman as you may value the services of the coal, by analysis, and the only trouble is, I think, that it is much more difficult to analyze the extra qualifications of the workingman than it is to analyze the extra value of the coal.

Mr. H. H. Suplee.—I think this question of the efficiency of workingmen depends partly upon wages, but very largely upon

* President of the Institution of Mechanical Engineers.

something which is not to be paid for in money, as Mr. Emerson has mentioned. An analogy to that I think has been found in some investigations recently made by Dr. J. M. Rice, in connection with the work of the Society for Educational Research, upon the operation of the public schools. These researches include what may be called additional or outside examinations of pupils—not to find out how much the pupils know, but to ascertain how well the pupils' teachers are doing their work—and some very curious results have been obtained. In some of the schools the pupils showed very high results, where apparently the teachers were not particularly able. In other cases where there were some very able teachers, the pupils did not show so well and the teachers did not get the same good result. The investigation was carried further and it was found that the real indication of success lay in the principals of the schools. It was found that an active, energetic principal could stimulate the teachers under him, and that energy was passed on to the pupils, and the result was a very high degree of efficiency. It was also found that where a change was made in the heads of schools this same remarkable change in results was observable. Now I think the same things bears true in a workshop. This brain element which is not drawn out by money, but which can be inspired by enthusiasm, is created by the foreman or the proprietor; and it is a fact which many of us know that there are shops in this country where the foreman and the managers are looked up to, admired and loved, by the workingmen, and in such shops good results are always obtained, and there are other shops in which the foreman or the manager lacks the power to interest the men in their work, and gets them disgruntled and himself disliked. It is not only the wage question which enters; it is the personal equation that also enters. With the right man over the men the trouble will disappear.

*Mr. Harrington Emerson.**—In reflecting on the intelligence of the employer of labor, of course, I was not reflecting on those gentlemen that belong to the American Society of Mechanical Engineers, or the related English Society, because we are supposed to know all about those things. But I should certainly say that I do consider the average manufacturer and the average employer of labor as knowing very little about his business or its details. Much experience from going into some of the very largest

* Author's closure, under the Rules.

shops in this country has shown me things that are simply unbelievable, things that wouldn't be credited if I were to tell them here. Those who have studied these matters know that in a shop organized so as to obtain the very best from men and machines, output is increased anywhere from thirty per cent. to 100 per cent.

Now those matters are not matters of opinion, but they are matters that have actually been demonstrated in some of the largest concerns in this country. The usual lack of efficiency probably because the man at the top has had his mind taken up in planning his big business, in planning the buildings and getting the general scheme running, and then when one comes to details it will be found that he has been paying some man a daily wage—perhaps a generous wage—to attend to some matter, and fifty per cent. is lost, because the man is not of sufficient intelligence to get out of the machine, or out of the combination what there ought to be in it. Quite recently in going through a large model shop, I investigated the abrasive wheel conditions, and found efficiencies as low as four per cent. I saw the remains of a cup emery-wheel. The whole cup had been worn off, but the wheel was put on another spindle to be used as a sharpening wheel, and worn down until the brass web was sticking out all around the circumference, and the wheel itself broken into five pieces, held only by the hub. When I took it off the spindle the man said, "Please bring it back soon because we need it." Now that is the kind of thing I have found in every single shop in which I have ever been, not the particular emery-wheel illustration, but things of that kind. A man can go through a shop with his eyes closed and he can guess at the efficiency of the shop simply by his ear alone.

In the machine shop I mention a thorough reform of abrasive wheel conditions will effect yearly economies in value of time saved alone of $10,000, and additional economies of increased and better production of $20,000.

No. 1041.*

CAST IRON, STRENGTH, COMPOSITION, SPECIFICATIONS.

BY W. J. KEEP, DETROIT, MICH.

(Member of the Society.)

The data which we have for this investigation consist of nineteen series of tests made for the committee on tests of the American Society of Mechanical Engineers in 1894-95, and of twelve series made in 1899-1901 by the committee on tests of the American Foundrymen's Association.

The former of the above series were made on pairs of test bars cast together varying in size from ½ inch square to 4 inches square, and the latter series on round bars of the same areas.

To compare such records of ½-inch to 4-inch test bars it is necessary to find the strength of a section ½ inch square by 12 inches long of each which is the greatest common divisor of all test bars. See Fig. 427.

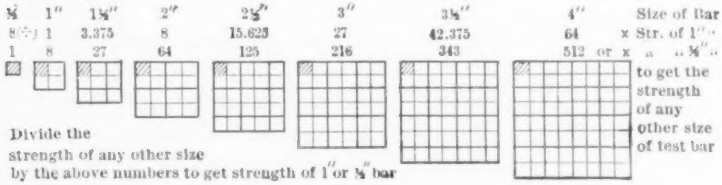


FIG. 427.

Tensile tests were made by the American Foundrymen's Association.

Complete chemical analyses were made of each series.

The American Society of Mechanical Engineers series are numbered and the American Foundrymen's Association series are lettered.

The American Society of Mechanical Engineers records are the average of two test bars of rectangular section cast together. The American Foundrymen's Association records are the average of all bars of the same area both square or round, in most cases the average of sixteen bars.

These strengths and the chemical composition of these bars are shown graphically in the Figs. 428 to 442.

* Presented at the Chicago meeting (May and June, 1904) of the American Society of Mechanical Engineers, and forming part of Volume XXV. of the *Transactions*.

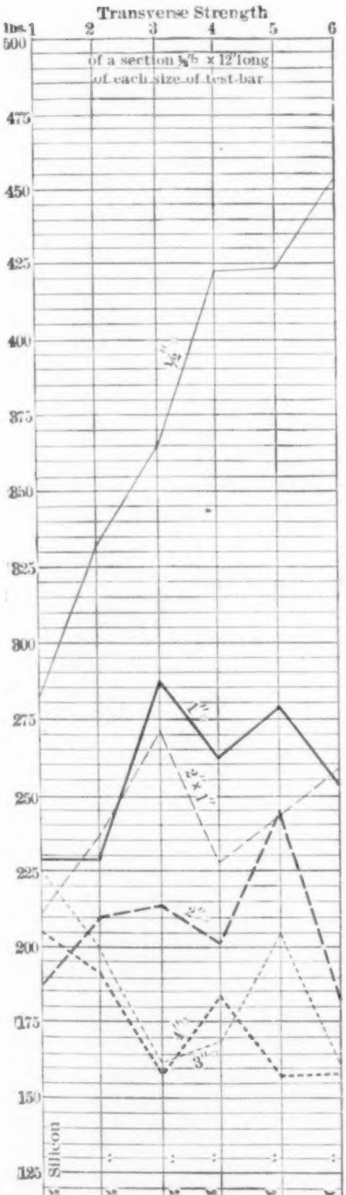


FIG. 428.

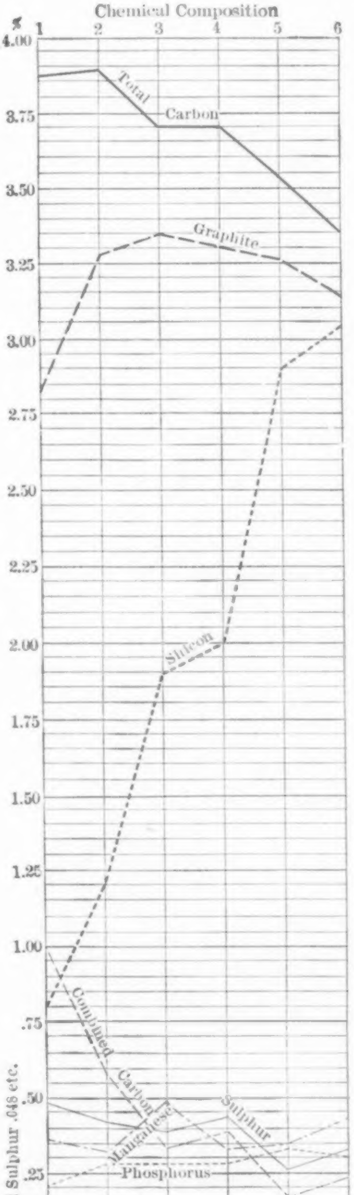


FIG. 429.

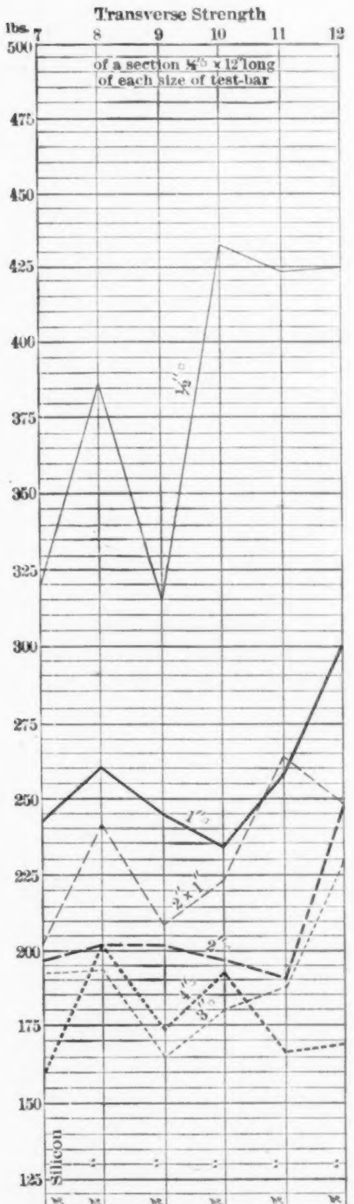


FIG. 430.

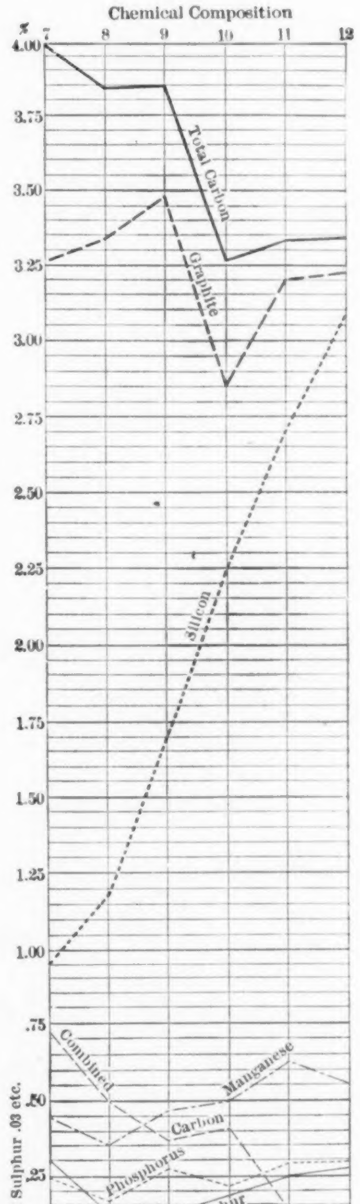


FIG. 431.

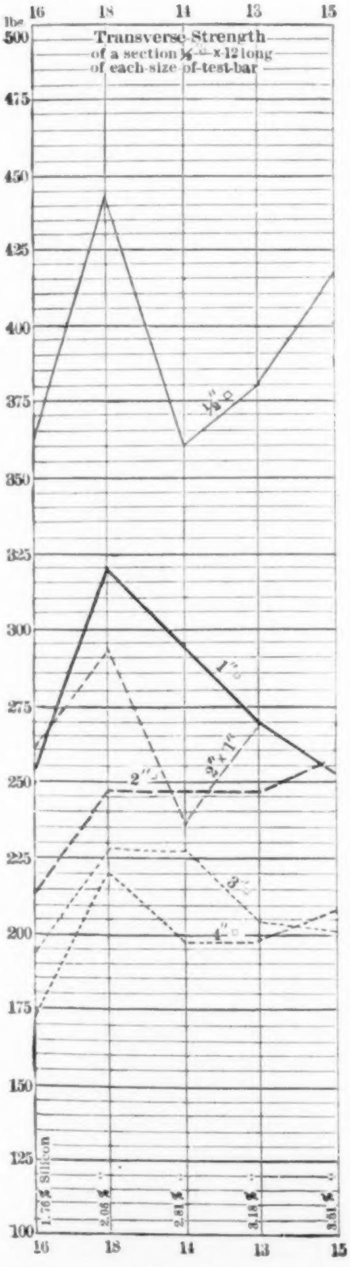


FIG. 432

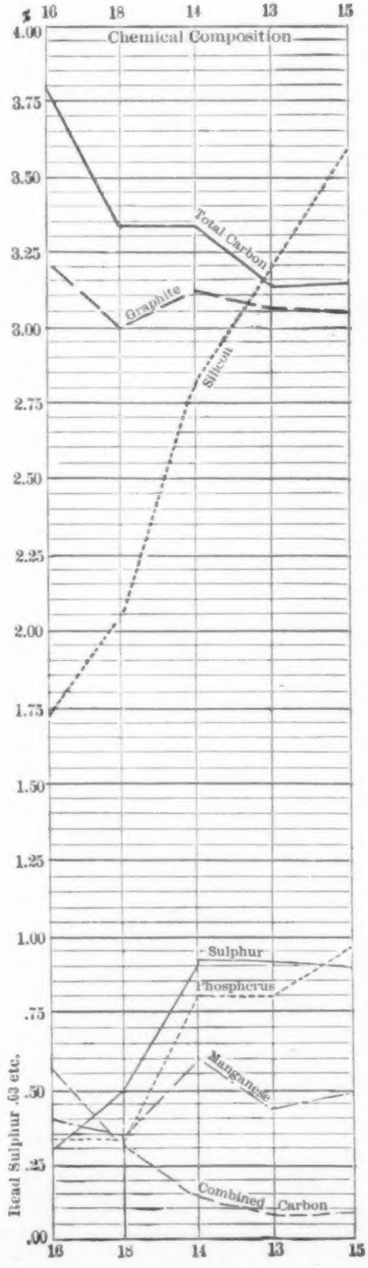


FIG. 433.

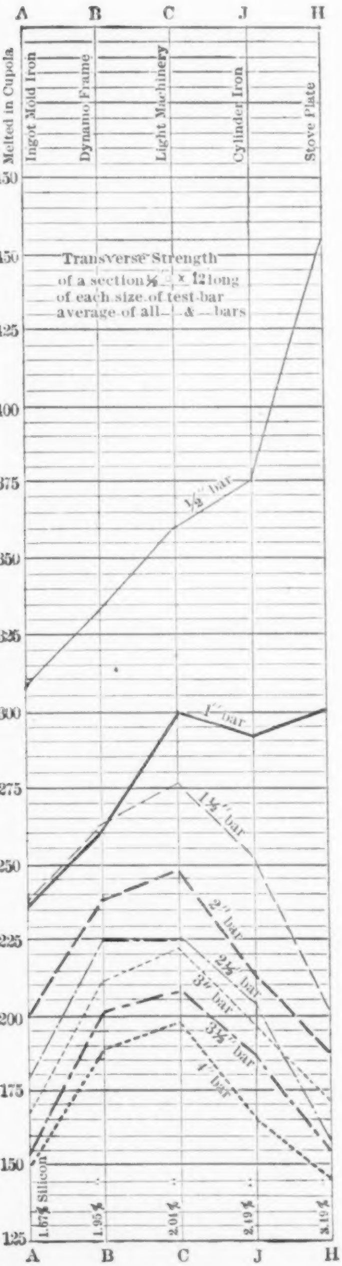


Fig. 434.

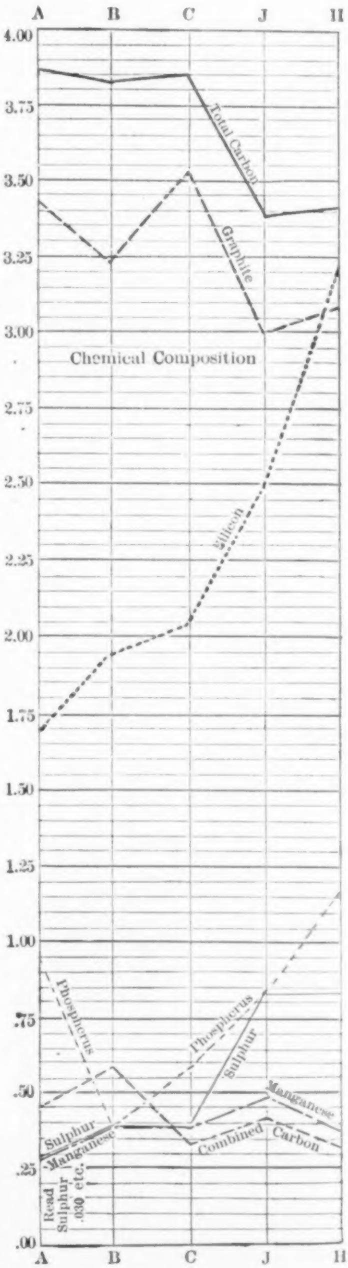


Fig. 435.

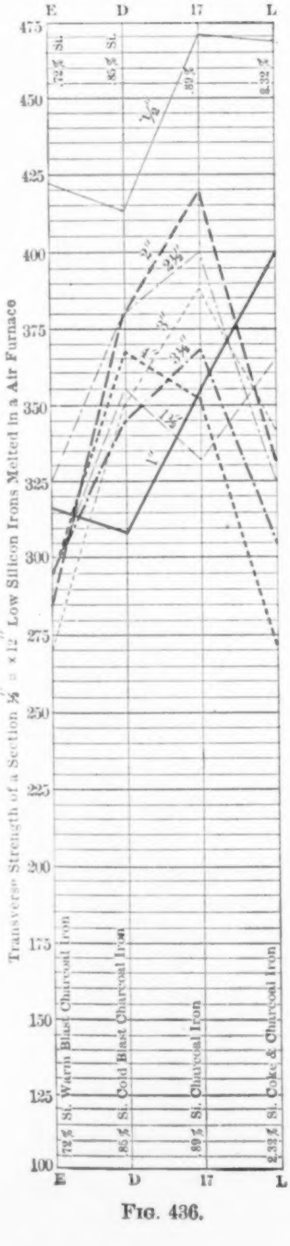


Fig. 436.

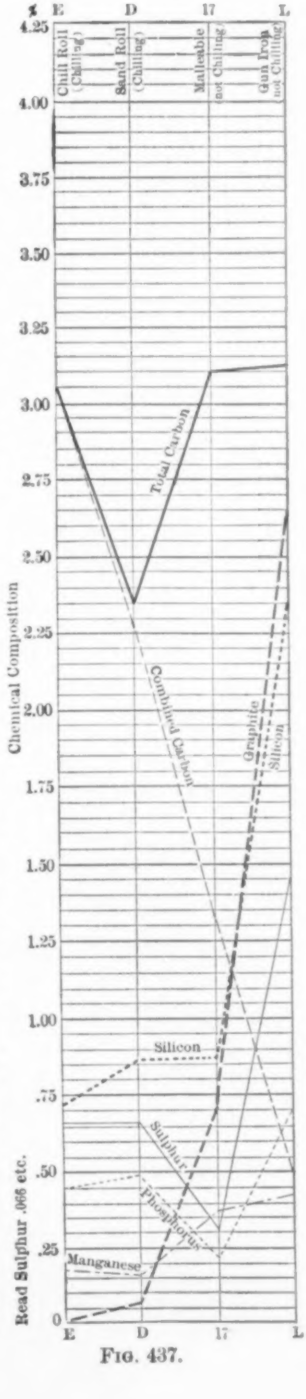


Fig. 437.

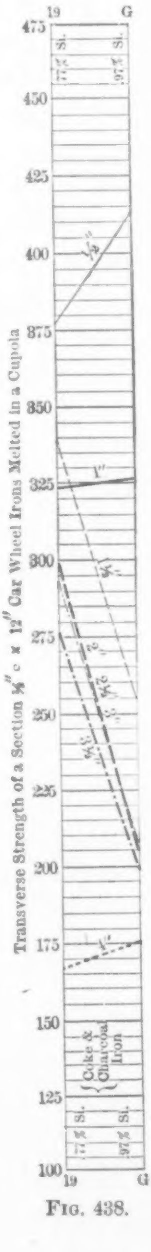


Fig. 438.

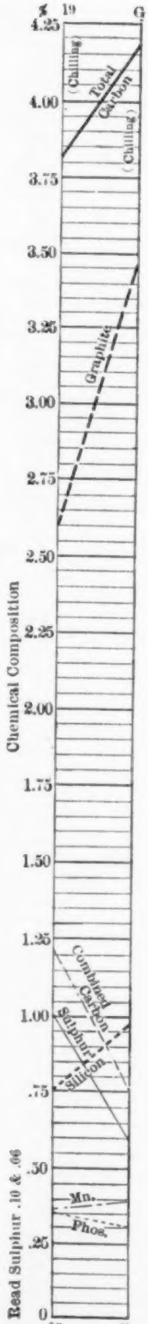


Fig. 439

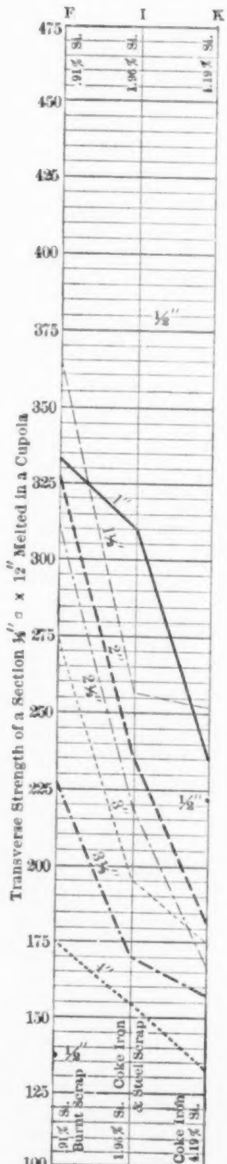


Fig. 440

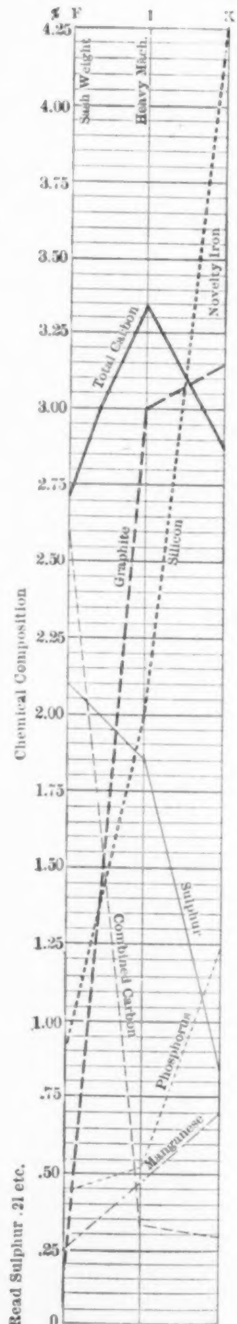


Fig. 441.

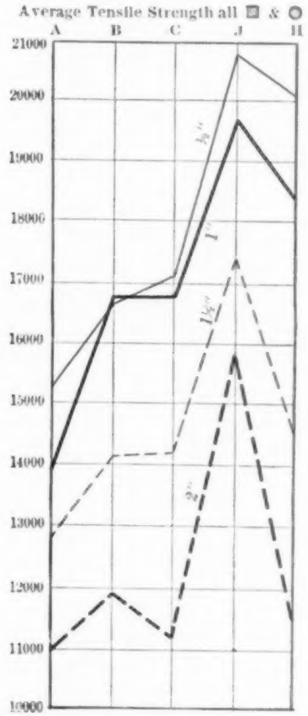


FIG 442.

PHYSICAL AND CHEMICAL SIMILARITY.

In the preceding figures, take the line or curve representing any chemical element and compare its variation with the variation of the line representing the strength of each size of test bar and note the apparent similarity. After completing the examination condense the results for the influence of each chemical element.

CHEMICAL COMPOSITION AND STRENGTH.—*Silicon.*—Up to 3.00 per cent. silicon increases the strength of small castings, such as ½-inch test bars. Up to 2.00 per cent. it increases the strength of 1-inch test bars.

In the air furnace casts *D, E* and *L*, ½-inch, 1-inch and 1½-inch test bars all increase in strength as silicon increases, and all other sizes increase in strength to .90 per cent. silicon.

In cupola mixtures with silicon over 1.00 per cent. test bars larger than 1 inch usually decrease in strength as silicon increases.

The tensile tests, Fig. 442, of all bars up to 2 inches and to 2.50 per cent. show an increase in strength, but a decrease for 3.00 per cent.

Total Carbon.—In gray iron, transverse strength decreases with increase of total carbon. The tensile strength shows no such uniformity.

In Series 1, Fig. 438, steel scrap was added to decrease total carbon and to increase strength, but this was not as strong as *B*, Fig. 434, with no scrap, though both have the same silicon.

Total carbon increases as silicon decreases because silicon changes combined to graphitic carbon, and some of this escapes as the metal cools.

The increase in strength and decrease in total carbon in test bars up to 2 inches is caused by the increase of silicon which removes brittleness.

Combined Carbon.—This always decreases as silicon increases with normal conditions. The transverse strength of ½-inch and 1-inch test bars gradually increased while combined carbon decreased, but in larger bars strength and combined carbon both decreased.

This was due to the slow cooling which increased the size of the grain. The average of all tensile tests shows an increase of strength as combined carbon decreases.

If it was the decrease of combined carbon which caused the decrease in strength in large test bars the smaller bars would not show the opposite result.

Analysis of each size of test bar often shows the same combined carbon in small and in large bars, but the small bars are invariably strong and the large bars weak due to slow cooling.

In Fig. 435, *B* has the lowest combined carbon and the greatest strength of the group. In *J* the strength drops in all sizes of test bars while combined carbon is slightly greater.

Comparing *B* with *I* (Fig. 439), while both have the same silicon, *I* has very much lower combined carbon and is very much stronger in the ½-inch and 1-inch bars; is about the same strength in the 1½-inch, 2-inch and 2½-inch bars, but the 3-inch, 3½-inch and 4-inch bars are much weaker. (Steel scrap added to *I* did not act as expected.)

Closing the grain and removing brittleness increases strength. Melting in an air furnace, Fig. 440, increases both strength and combined carbon.

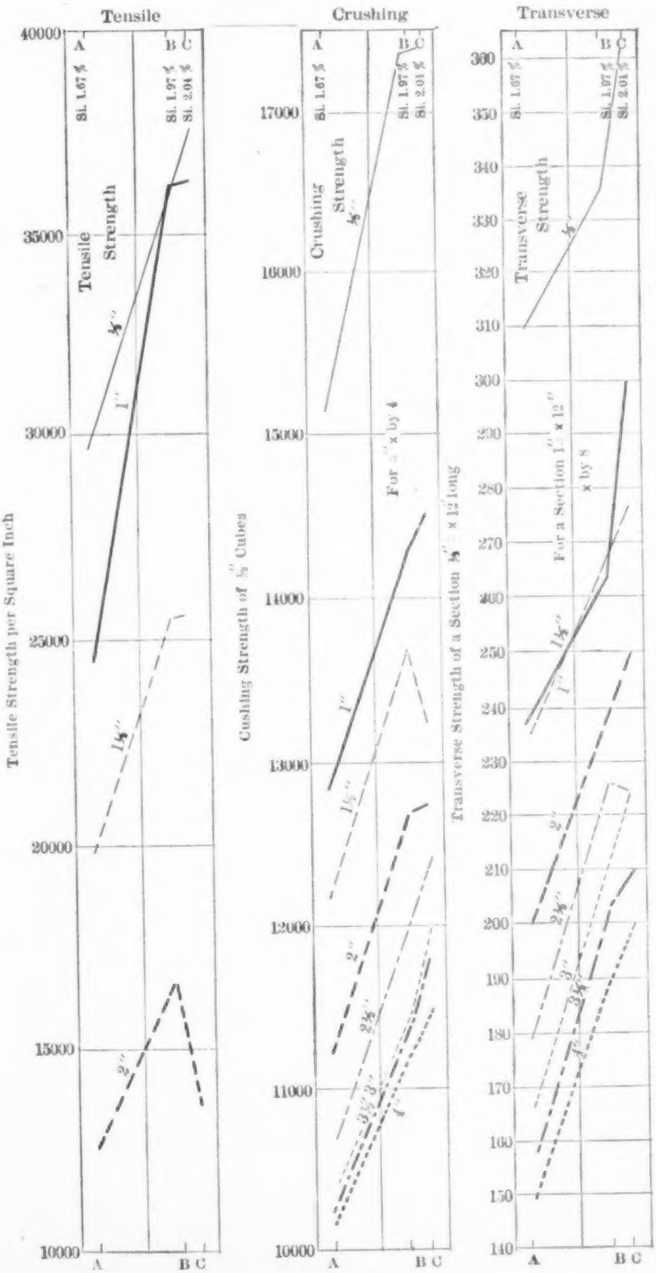
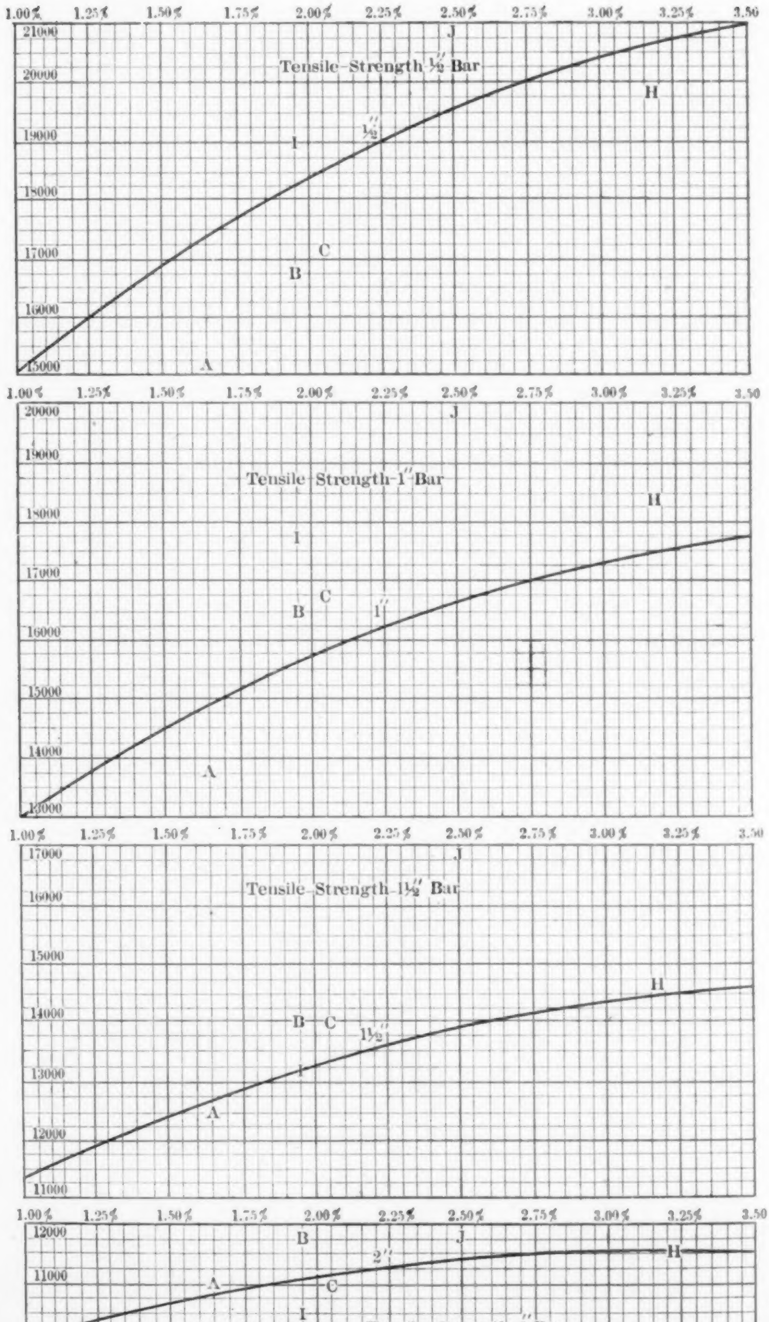


Fig. 443. Fig. 444. Fig. 445.



Tensile Strength ½" Bar
Tensile Strength 1" Bar
Tensile Strength 1½" Bar
Tensile Strength 2" Bar
1.00% 1.25% 1.50% 1.75% 2.00% 2.25% 2.50% 2.75% 3.00% 3.25% 3.50
A B C I J H

FIG. 446.

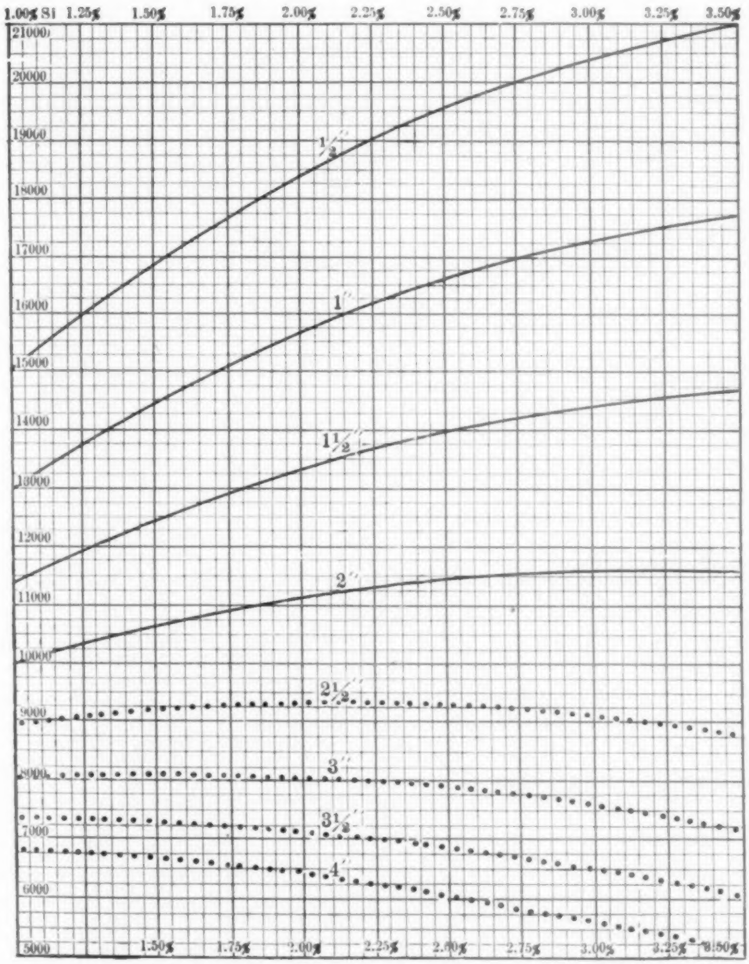


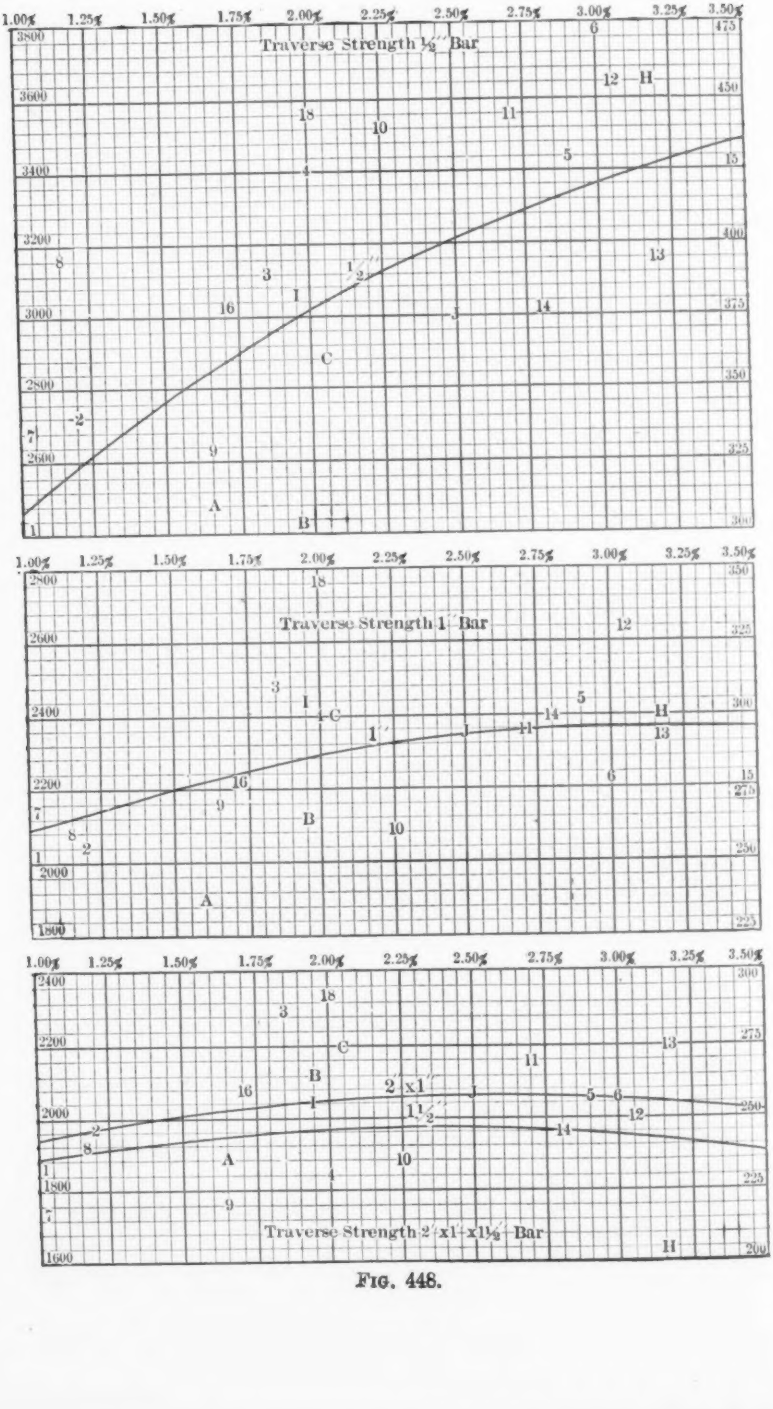
FIG. 447.—KEEP'S TENSILE STRENGTH CHART.

Approximate relation of strength to size of casting and to percentage of silicon (Table gives the strength per square inch.)

Graphitic Carbon.—The quantity in any casting is the difference between the total and the combined carbon. In these series there is no uniformity between the percentage of graphite and the strength.

Phosphorus.—In all of these series phosphorus generally increases as silicon increases.

While the tensile tests, Fig. 442, show an increase of strength



1.00% 1.25% 1.50% 1.75% 2.00% 2.25% 2.50% 2.75% 3.00% 3.25% 3.50%
Traverse Strength ½″ Bar
3800
3600
3400
3200
3000
2800
2600
475
450
425
400
375
350
325
300
Traverse Strength 1″ Bar
2800
2600
2400
2200
2000
1800
350
325
300
275
250
225
Traverse Strength 2″ x 1″ x 1½″ Bar
2400
2200
2000
1800
1600
300
275
250
225
200

Fig. 448.

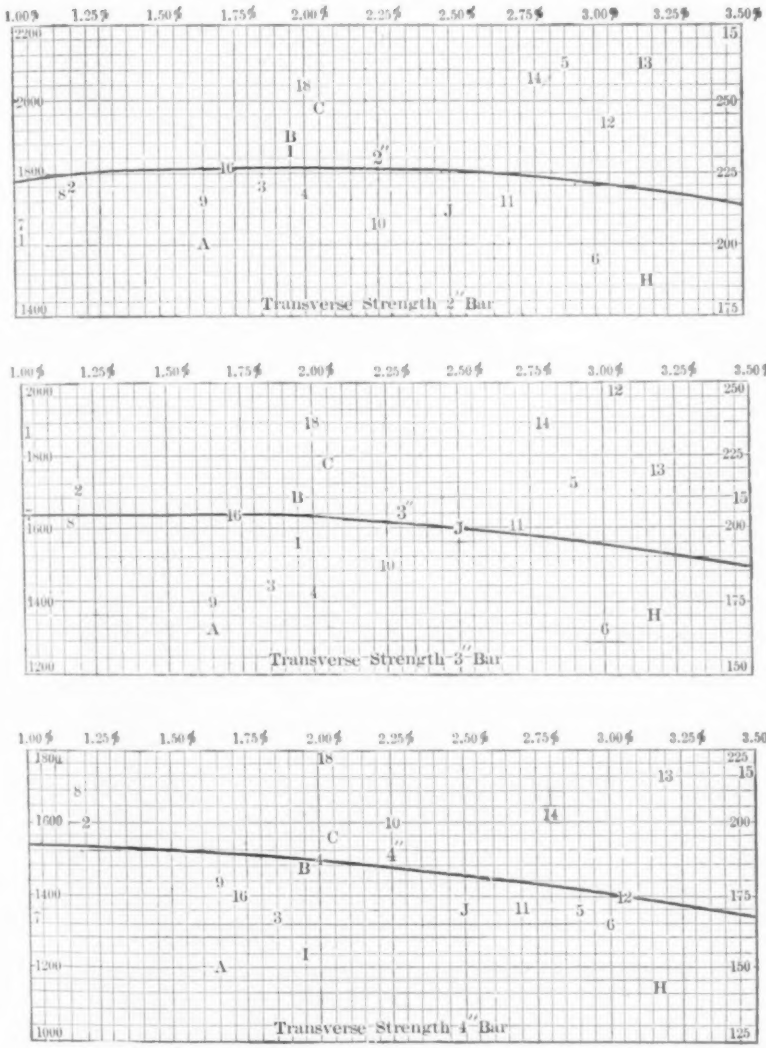


Fig. 448—Cont'd.

with an increase of phosphorus, yet the transverse tests, especially Figs. 432 to 435, seem to show that phosphorus reduces strength. This is also general shop experience.

Sulphur.—There is not in these tests enough uniformity between the percentage of sulphur and the strength to show any decided influence, but the indication is that sulphur decreases

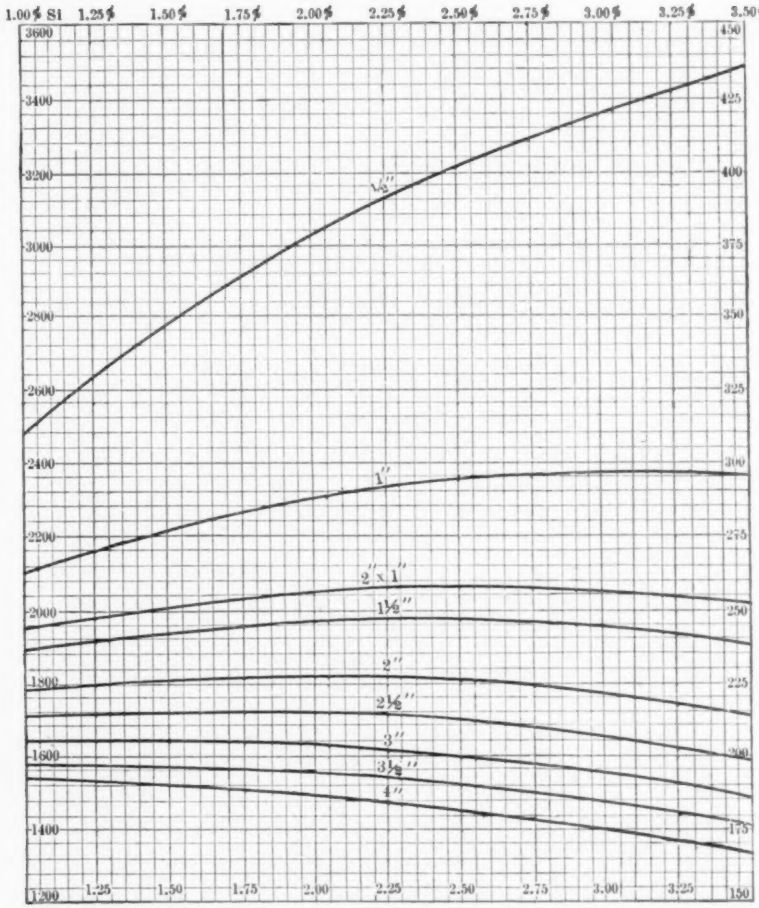


FIG. 449.—KEEP'S TRANSVERSE STRENGTH CHART.

Approximate relation of strength to size of casting and to percentage of silicon.
(Table gives the strength of a section of each test bar 1″ □ × 12″ long.)

strength. In some cases sulphur might add to strength by causing the grain to be closer.

Manganese. — The percentage is too nearly the same in these series to show any influence on strength.

By comparing strengths and chemical composition of the irons nearest alike, as 3, 9, 16, *A*, or 5, 18, *B*, *C*, *I*, or 13, *H*, or 6, 12, with all chemical elements nearly alike, and no scrap, but with quite different strengths, it is very evident that strength is de-

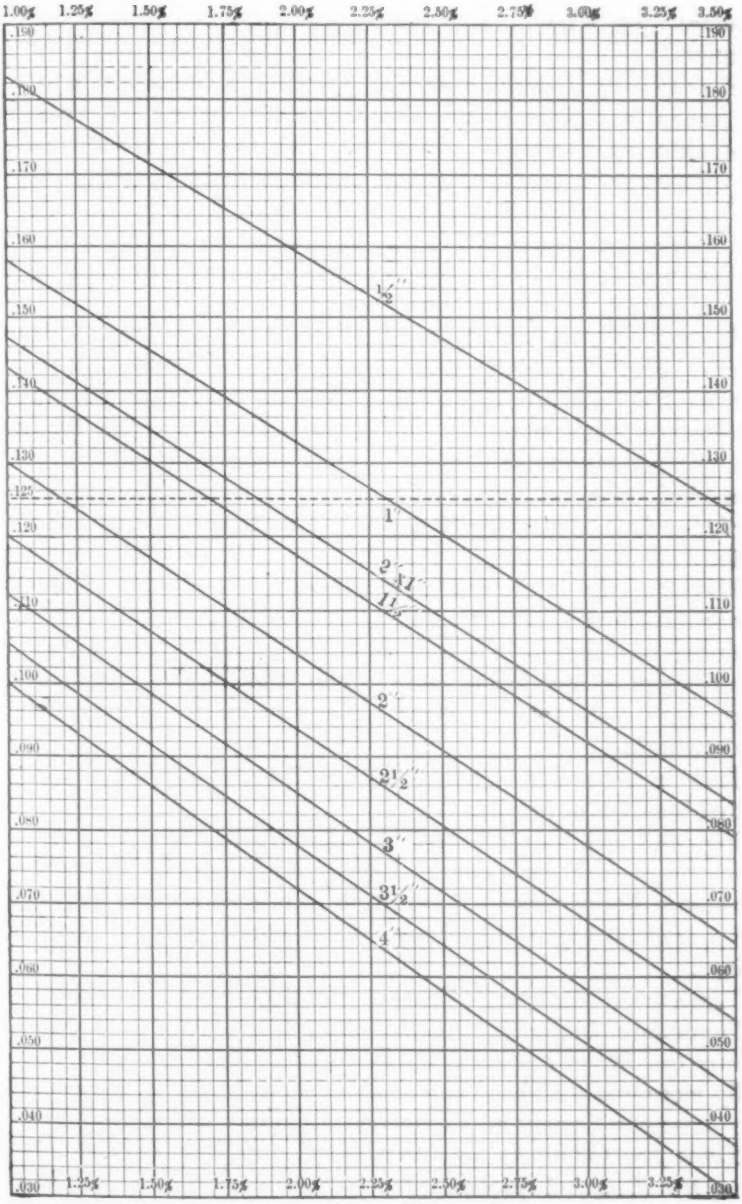


FIG. 450.—KEEP'S SHRINKAGE CHART.
Approximate relation of shrinkage to size of casting and to percentage of silicon.

pendent upon something outside of the ordinary chemical composition.

Slow Cooling Decreases Strength by making the grain of a casting coarse and more open. The larger the casting the weaker it becomes per square inch of section. The weakness is not caused by a decrease in combined carbon because a complete analysis of each size of test bar (*Transactions*, American Society of Mechanical Engineers, Voume XVI., p. 1100) shows the same combined carbon in all sizes of many series, but in all cases the strength per unit of section decreased as the size increased.

Strength of Any Size of Test Bar cannot be Calculated by Any Mathematical Formula from the Measured Strength of Another Size, because the grain changes by slow cooling. Such strength must be obtained by a graphic chart. Fig. 443 shows the average tensile strength per square inch, Fig. 444 the average crushing strength of a ½-inch cube, and Fig. 445 the average transverse strength of a section 1 inch square by 12 inches, of each size of test bar of series *A*, *B* and *C*.

The similarity in the diagrams of each of these three kinds of strengths shows that a graphic chart should show this general character of diagram. Fig. 446 shows the average tensile strength of each size of test bar of each of the American Foundrymen's Association tests and also gives a line showing the average tensile strength.

Tensile Strength Chart.—Fig. 447 shows this chart. The dotted lines are estimated.

Fig. 448 and Fig. 448 (cont'd.) show the average transverse strengths of each American Society of Mechanical Engineers and American Foundrymen's Association tests and a curve showing the average strength of each size of test bar of each variation in silicon.

Transverse Strength Chart.—Fig. 449 shows these curves.

Shrinkage Chart for approximating the percentage of silicon in any test bar or casting, Fig. 450 is constructed from the carefully measured shrinkage and analyses of each size of test bar of the American Society of Mechanical Engineers series.

Table for Obtaining the Strength of Any Size of Test Bar from the Measured Strength of a Standard Test Bar.—Table I is calculated from chart, Fig. 449, for a standard 1-inch square test bar. Measure the shrinkage per foot of the standard test bar, then on the shrinkage chart, Fig. 450, find this shrinkage on

TABLE I.

KEEP'S TABLE FOR APPROXIMATE TRANSVERSE STRENGTH.

Per cent. of Silicon..	1.00%	1.25%	1.50%	1.75%	2.00%	2.25%	2.50%	2.75%	3.00%	3.25%	3.50%
SIZE OF TEST BAR:											
½ inch sq....	1.179 .1473	1.246 .1558	1.253 .1567	1.287 .1610	1.318 .1648	1.346 .1683	1.370 .1713	1.394 .1743	1.418 .1772	1.443 .1804	1.471 .1839
1 inch sq.	1.	1.	1.	1.	1.	1.	1.	1.	1.	1.	1.
2 × 1 in. sq.	.9286 1.857	.9188 1.838	.9095 1.819	.8982 1.796	.8932 1.787	.8860 1.772	.8787 1.757	.8750 1.750	.8587 1.717	.8586 1.717	.8520 1.704
1½ in. sq....	.8976 3.029	.8863 2.991	.8733 2.947	.8629 2.912	.8562 2.890	.8473 2.860	.8383 2.829	.8326 2.810	.8228 2.777	.8122 2.741	.8055 3.141
2 in. sq.....	.8429 6.743	.8306 6.645	.8167 6.534	.8009 6.407	.7908 6.327	.7806 6.245	.7681 6.145	.7585 6.068	.7447 5.958	.7342 5.873	.7225 5.780
2½ in. sq....	.8117 12.68	.7935 12.40	.7783 12.16	.7611 11.89	.7473 11.68	.7365 11.49	.7213 11.27	.7098 11.09	.6962 10.88	.6857 10.71	.6701 10.47
3 in. sq....	.7833 21.64	.7610 20.55	.7421 20.04	.7235 19.53	.7102 19.18	.6968 18.81	.6808 18.38	.6695 18.08	.6540 17.66	.6413 17.32	.6300 17.01
3½ in. sq ...	.7524 32.26	.7309 31.34	.7104 30.46	.6925 29.69	.6776 29.05	.6624 28.40	.6468 27.73	.6356 27.25	.6224 26.68	.6097 25.84	.5962 25.56
4 in. sq.....	.7654 46.78	.7100 45.44	.6900 44.16	.6681 42.76	.6493 41.55	.6344 40.60	.6191 39.63	.6059 38.78	.5907 37.81	.5781 36.99	.5646 36.10

the left-hand margin and follow horizontally until you intersect the line of the measured test bar. Follow the vertical line at the intersection to the top of the chart, and you find the percentage of silicon that is expected to produce that shrinkage. Find this same percentage at the top of Table I, and follow down to the size of test bar that you wish the strength of. If you wish the actual strength use the lower figures as a multiplier of the measured strength of the standard 1-inch bar. If you wish the strength of a section 1 inch square by 12 inches long of the required test bar use the upper number to multiply by.

If you have the strength of any size of test bar other than a 1-inch bar, and know the silicon percentage, divide such strength by the lower number for the bar, or if you have the strength of a section of the required test bar 1 inch square by 12 inches long, divide by the upper number, and the result in either case is the strength of the standard 1-inch bar.

To Find the Strength of Any Casting.—Divide the cubic contents of a casting by the square inches of cooling surface, and the quotient is the cooling ratio. If the casting has a large flat surface the edges may be neglected; for example, a casting 1 inch thick and 24 inches square. A strip 1 inch wide and 24 inches long would have 24 cubic inches contents and 48 square inches

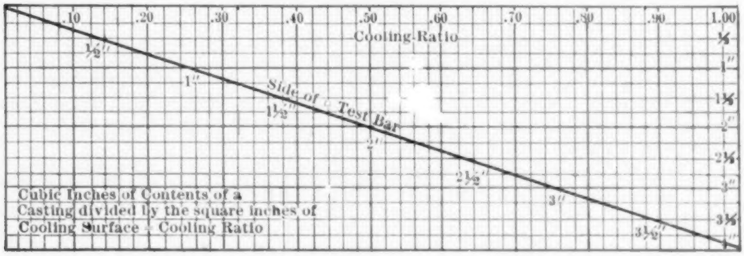


FIG. 451.

of cooling surface. 24 ÷ 48 = .5 ratio. Find this ratio at the top of the chart, Fig. 451, and follow down to the diagonal, and we find that a 2-inch square test bar represents the strength of the casting.

With the shrinkage of a standard 1-inch test bar, cast at the same time as the casting, find on the shrinkage chart the percentage of silicon in the casting, then in Table I find the upper multiplier for a 2-inch test bar. This multiplied by the measured strength of the standard test bar gives the strength of a section of the casting 1 inch square and 12 inches long.

PROPOSED SPECIFICATIONS FOR CAST IRON.

At the appointment of a committee by the International Association for Testing Materials at Zurich in 1895, the charge was: "On the basis of existing specifications to seek methods and means for the introduction of international specifications for testing and inspecting iron and steel of all kinds." The secretary of the American section of the committee on cast iron says that the "committees began to collect information on existing methods and to formulate specifications based thereon as far as possible, the final results being intended to represent the best American practice at the present time."

The following are submitted as desirable by the various committees:

GENERAL GRAY IRON CASTINGS AND METHOD OF TESTING.—*Chemical Properties.*

Light castings sulphur not over		0.08 per cent.
Medium " " " "		0.10 " "
Heavy " " " "		0.12 " "

Definition.—Light castings are those less than ½ inch thick. Heavy castings more than 2 inches thick. Medium all between.

Physical Properties. Transverse Test.—The minimum breaking strength of the "Arbitration Bar" (1¼ inches diameter) under transverse load shall not be under

Light castings	2,500 lbs.
Medium "	2,900 "
Heavy "	3,300 "

In no case shall the deflection be under .10 of an inch.

Tensile Test not less than

Light castings	18,000 lbs. per sq. in.
Medium "	21,000 " " " "
Heavy "	24,000 " " " "

Two sets of round bars 1¼ inches diameter shall be cast from each heat, one set from the first and the other set from the last iron going into the castings. The transverse test shall be made on all the bars cast with supports 12 inches apart, load applied at the middle. One bar of every two of each set made must meet requirements.

AMERICAN FOUNDRYMEN'S ASSOCIATION proposed specification. Light and Medium weight castings silicon 1.75 per cent. and up, test bars 1½ inches diameter.

Heavy castings silicon 1.50 to 2.00 per cent., test bars 2 inches diameter.

Chilling irons silicon below 1.00 per cent., test bar 2½ inches diameter.

No specification for strength.

PIPE CASTINGS, no chemical specification given.

Physical Test.—Test bar 2 inches wide, 1 inch deep, supports 24 inches apart and loaded at center. For pipe 12 inches diameter and less, breaking load 1,900 pounds with not less than 30-inch deflection.

For pipe larger than 12 inches, load 2,000 pounds with deflection not less than .32 inch. The test shall be based upon the average result of three test bars.

LOCOMOTIVE CYLINDERS.—*Chemical Properties.*

Silicon	from	1.25 to 1.75 per cent.
Phosphorus	not over	.90 per cent.
Sulphur	" "	.10 " "

Physical Properties.—"Arbitration test bar," $1\frac{1}{4}$ inches diameter, supports 12 inches apart, strength not less than 2,700 pounds, deflection not less than .08 inch. One test bar for each cylinder. Acceptance or rejection in case of dispute based on chemical analysis.

MALLEABLE CASTINGS.—*Chemical Properties.*—Sulphur not over .06, phosphorus not over .225.

Physical Properties.—Standard test bar 1 inch square, supports 12 inches apart, transverse strength after annealing not less than 3,000 pounds, deflection at least $\frac{1}{2}$ inch. Tensile strength, the same size of bar, not less than 42,000 pounds per square inch.

"EXISTING METHODS" AND SPECIFICATIONS.

The reply to letters to leading founders and chemists was generally that they used no specifications.

THE UNITED ENGINEERING AND FOUNDRY COMPANY, comprising many of the largest foundries in the Pittsburg district, says: "We watch our silicons and sulphurs pretty carefully for the ordinary run of castings, and when we desire castings of high strength we make a mixture from scrap and pig that, when melted in an air fuurnace, will give us a silicon of about 1.50 per cent."

PHILADELPHIA AND READING RAILWAY COMPANY.—"With 50 per cent. of pig iron as per specification and 50 per cent. of good scrap with ferro manganese in the ladle, we got a very tough, close-grained iron which turns up almost like steel. We get the best results by the combination of analysis from the castings themselves combined with the appearance and character of the fracture, and we avoid test bars owing to the difficulty of having them represent the general condition of the castings. We used a test bar 1.13 inches diameter, 12 inches between supports. Our former specifications were roughly.

Medium Iron, engine cylinders, gears, etc. Silicon 1.40 to 2.00 per cent. Sulphur less than .085 per cent. Phosphorus less than .60 per cent. Manganese less than .70 per cent. Transverse strength about 2,400 pounds per square inch."

"*Soft Iron*, general car and roadway use. Silicon 2.00 to 2.80

per cent. Other elements same as medium iron. Transverse strength 2,000 pounds."

"*For Brake-shoes* and other castings for frictional wear. Silicon 2.00 to 2.50 per cent. Sulphur less than .15 per cent. Phosphorus less than 0.70 per cent. Manganese less than 0.70 per cent. Transverse strength 2,900 pounds."

J. I. CASE THRESHING MACHINE COMPANY SPECIFICATIONS and used by a large number of Western founders, and published by their chemist, Mr. W. G. Scott, is as follows:

Close Hard Iron for air and ammonia compressors, H. P. cylinders, H. P. valves, etc. Silicon 1.20 to 1.60 per cent. (below too hard, above porous, unless much scrap is used). Sulphur less than .095 per cent. Phosphorus below 0.70 per cent. (for chill .30 per cent.). Manganese below .70 per cent. (higher for chill). Transverse strength, test bar 1 inch square by 12 inches cast in yokes, 2,400 pounds. Tensile strength per square inch 22,000. Shrinkage (bar 1 inch square by 12 inches), not more than .161 inch. Chill in yokes below .25 inch.

Medium Iron for engine cylinders, gears, pinions, etc. Silicon 1.40 to 2.00 per cent. (1.50 best for gears). Sulphur less than .085 per cent. (best .075 to .080). Phosphorus below .70 per cent. Manganese below .70 per cent. Transverse strength 2,200, tensile 2,000. Shrinkage not more than .154 inch, chill .15 inch.

Soft Iron for pulleys, small castings and agricultural work. Silicon 2.20 to 2.80 per cent. (below too hard, above weak for large castings, 2.40 a good average). Sulphur less than .085. Phosphorus below .70 (1.00 for stove plate). Manganese below .70 per cent. Transverse strength 2,000. Tensile 18,000. Shrinkage .141 inch. Chill .05 inch.

THE LORAIN FOUNDRY COMPANY.—Mr. Oliver Phelps, former general manager, gave me the following: "The efforts of the Lorain foundry were devoted specially to large and heavy castings. Size of test bar 2 inches wide, 1 inch deep and 24 inches between supports. Three test bars.

Hard Iron, for compressor cylinders, valves and high-pressure work. Silicon 1.20 to 1.50 per cent. Sulphur under .09. Phosphorus .35 to .60. Manganese .50 to .80. Transverse strength 2,600 pounds per square inch. Tensile 24,000 pounds per square inch.

Medium Iron for general work. Silicon 1.50 to 2.00 per cent. Sulphur under .08 per cent. Phosphorus .35 to .60. Manganese

.50 to .80. Transverse strength 2,400 pounds per square inch. Tensile 23,000. The above are cupola melts with best Connellsville coke. Limestone flux.

In some cases for density we place in the ladle ½ to 1 per cent. of aluminum. For castings of over 15 tons we sometimes mixed air furnace and cupola iron, gaining nearly 30 per cent. increase in strength in both tests. I think our high tests were, however, largely due to Northern lake ore irons, not over 15 per cent. of our mixture being made with Southern iron.

Chemical Work Castings (cupola). Silicon 1.10 to 1.35. Sulphur under .07. Phosphorus under .25. Manganese .40 to .60 per cent.

Air Furnace Iron.—For Omaha and Pittsburg pumps 15 to 20 millions capacity, the average thickness of metal 2¼ inches, the several pieces weighing 12,000 to 29,000 pounds each. The analysis was T. C. 3.20, G. C. 2.19, C. C. 1.01, Si. 1.30, S. 0.085, Mn. 0.37.

This iron was melted in an air furnace with gas coke running about 1.50 per cent. sulphur, about 1 pound coal to 3 pounds iron. Time of heat about seven hours."

English Practice.—Prof. Thomas Turner says: "For transverse test the common test adopted by iron founders is breaking a bar 3 feet long by 2 inches deep and 1 inch broad. However, many shapes and sizes of test bar have been adopted, and for scientific purposes the results so obtained are converted by calculation into values for a bar one foot long and one inch square."

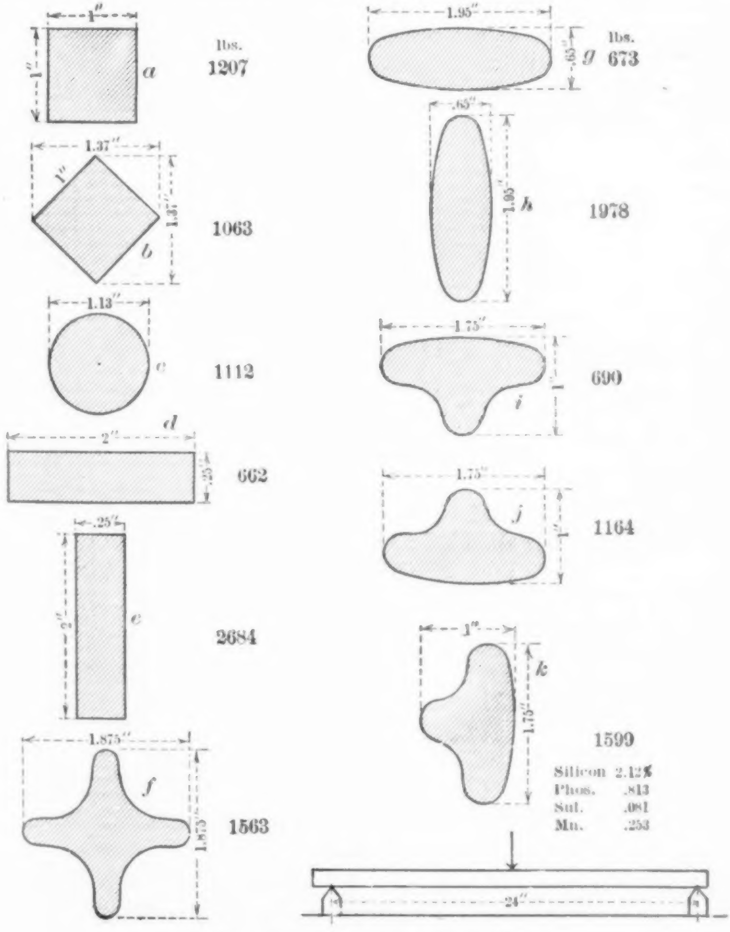
Comments on Specifications.

Chemical Specification.—The chemical properties for each kind of casting should be specified. A small variation in silicon will make castings either too hard or too porous. The general founder should be instructed on these questions. The sulphurs of the committee for general gray iron casting are too high. The practice of J. I. Case T. M. Co. and Lorain Foundry Company should be followed closely.

A chemical specification is of the utmost importance. The Lorain air furnace iron has silicon 1.30 per cent., the grain is close and strength of a 1 inch square bar is 3,900. The American Foundrymen's Association (cast *L.*) furnace iron has silicon 2.35 per cent., which would make the grain too open and would cause

spongy iron. To insure the density required for peculiar work the chemical composition must be specified.

TEST BARS.—*Best Size*—Very few realize the influence on strength due to a slight change in proportion of test bar. The re-



The area of each fracture is 1 square inch.

FIG. 452.

sults in Fig. 452 were given to me for this paper by the Dodge Manufacturing Company of Mishawaka, Ind. The fracture of each test bar was exactly one square inch area. The supports were 24 inches apart; strength the average of four bars. The

analysis of the iron was: Silicon 2.12. Phosphorus .813. Sulphur .081. Manganese .253.

Fig. 453 is a graphic chart of all of the preceding specifications reduced to terms of a bar 1 inch square by 12 inches long. There are four sizes of test bars, viz.:

1 inch square by J. I. Case and committee on malleables.

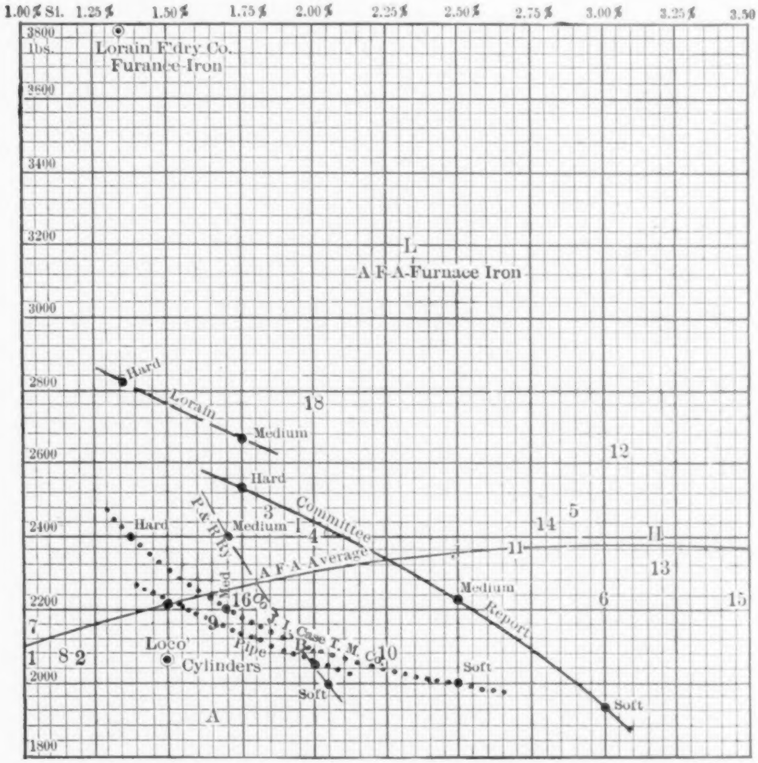


FIG. 453.

1.13 inches round by Philadelphia & Reading Railway.

1.25 inches round suggested by committee on general castings and for locomotive cylinders.

2 inches by 1 inch by 24 inches by committee on Pipe and Lorain Foundry Company and in England.

Tensile strength is always given in terms per square inch of area. The test bar for tensile test should be cast 1.13 inches diameter, parallel for 2 inches at centre, then gradually increas-

ing in size to give taper ends for a firm hold with the grips of the machine. The bar should be tested just as cast, which would give, without calculation, the tensile strength per square inch of a test bar cast and tested with a square inch section.

If a transverse test bar is cast 1 inch square and tested with supports 12 inches apart, the result is comparable with the tensile test because the area and the grain is the same.

The test bar 1.13 inches diameter has the same grain as the 1 inch square, and the strength is therefore comparable with 1

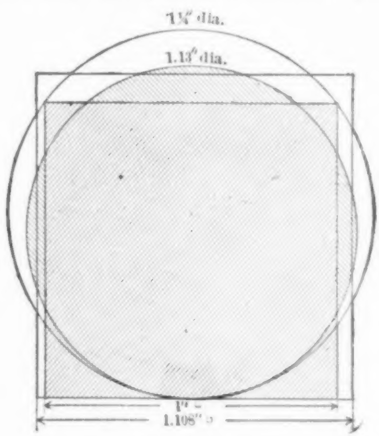


FIG. 454.

inch square, or with the tensile strength of a bar cast 1 square inch in section.

The "*Arbitration Test Bar*," 1.25 inches diameter. It cannot be used to arbitrate a dispute, because it is directed that the founder break all the bars, and no others can be made.

It was not intended in the original instructions that any committee should propose a new test, but to "base suggestions on existing methods." The record of a 1¼-inch round bar cannot be compared with any existing published records.

A round test bar 1.13 inches diameter equals a 1-inch square bar, equals area 1.00 square inch. See Fig. 454. A test bar 1.25 inches diameter equals a bar 1.108 inches square, equals area 1.227 square inches.

Has such a test bar any advantage that would warrant the discarding of all previous records?

It is not practical for a founder to reduce the record of a 1.25-

inch round bar to that of a bar with 1 square inch area, cast from the same iron, by any mathematical formula.

It is not possible at present to realize what the strength of such a test bar indicates.

Take, for example, the strength of a $1\frac{1}{4}$-inch bar at 2,700 pounds for locomotive cylinders.

The transverse strength of a 1-inch square section of a $1\frac{1}{4}$-inch round or a 1.108-inch square bar of 2,700 pounds by the formula

$$= \frac{2,700 \times 1^3}{1.108^3} = 1,985 \text{ pounds.}$$

But a 1-inch square section of a $1\frac{1}{4}$-inch round test bar, with 1.50 per cent. silicon, is 85 pounds weaker than a bar from the same iron cast 1 inch square.

1,985 + 85 = 2,070 pounds. (By Table I with 1.50 per cent. silicon this 1-inch square bar equals 2,060 pounds.)

Did the committee mean to prescribe such a low strength for locomotive cylinders?

The strength for general castings becomes

	1.25" dia.		Section 1" □		Loss for Slow Cooling		Bar Cast 1" □
Hard Iron	3,300 lbs.	=	2,438 lbs.	+	95 lbs.	=	2,533 lbs.
Medium	2,900 "	=	2,127 "	+	105 "	=	2,232 "
Soft	2,500 "	=	1,816 "	+	120 "	=	1,936 "

This is so near the J. I. Case T. M. Co. specification of 2,400, 2,200, 2,000 that it would seem better to adopt a specification in such general use.

For cast pipe the bar 2 inches wide, 1 inch deep and 24 inches between supports gives a record which is exactly the same as an average section of that bar 1 inch square by 12 inches long, but the larger bar looses considerable from slow cooling.

2 × 1 × 24	Section 1" □ × 12"	Lost	If Cast 1" □
2,000 =	2,000 +	227 =	2,227 lbs.
1,900 =	1,900 +	189 =	2,089 "

An average of eight test bars of series 18 of the English size 3 feet long 2 inches deep and 1 inch wide was 3251 pounds; the strength of a section 1 foot long and 1 inch square would be 2438 pounds.

The average of eight such bars poured from the same ladle and tested 1 inch deep and 2 inches wide was 1497 pounds; the strength of a section 1 foot long and 1 inch square would be 2246 pounds.

A test bar 2 inches by 1 inch whether 1, 2, or 3 feet long has a

sectional area of two square inches, but the strength per square inch of any of these bars whether tested the narrow or wide side down is not one-half the breaking load, but is the calculated strength of a section one foot long and one inch square.

Strength in Specifications.—It will be noticed from all figures, Nos. 428 to 449, and from the average American Foundrymen's Association diagrams, that as silicon is increased the strength of heavy castings decreases, but that the strength of a 1-inch square test bar from the same iron increases while in all specifications all heavy and medium castings and also the 1-inch square test bar must be strongest for low silicon. The chemical composition will not account for the extra strength. The Lorain Foundry Company gets it by using special lake ore iron. Some do so by adding steel scrap, and others by using charcoal and other irons with peculiar qualities. There are some pig irons that when mixed with other irons in the cupola give the desired strength and still keep the silicon at 1.30 per cent. Much less expensive irons are needed for medium iron, and quite low-priced irons can be used for soft castings. This is entirely independent of the chemical composition. For example, Series 18, Figs. 432 and 433, has silicon right for medium iron, but it has strength for the heaviest castings. The cupola charge was 500 pounds each, "Swede" (plain), "Pulaski" (No. 2), "Princess" (No. 2), "Kemble" (No. 2), 1,800 pounds scrap, 200 pounds cast iron borings. The test bars were cast at the middle of the heat. A test piece turned to $1\frac{1}{4}$ inches gave a tensile test of 29,040 pounds per square inch. Analysis of test bar was: T. C. 3.33, G. C. 2.83, C. C. .52, Si. 2.05, P. .342, S. .052, Mn. .354. Aside from its favorable composition the very high strength was due to the careful selection of pig irons, but most to the closing the grain with cast iron borings.

If the specifications of J. I. Case T. M. Co. be adopted for ordinary foundry iron, it might be well to specify a high grade cupola casting for extra heavy work, and take for such the Lorain strengths 2,800 and 2,600.

For the highest grade of air furnace iron we might take the Lorain for the strongest, and the American Foundrymen's Association, Series *L*, for the highest silicon allowed for medium weights. Strengths 3,800 and 3,200.

Square and Round Test Bars Cast Flat or on End.—As ordinary castings have flat surfaces and are cast flat, it would seem most natural to use a square test bar cast flat.

The committee recommended a piece of 10-inch water pipe for a flask. It is impossible to ram a vertical mould with a test bar extending upward without moving the pattern and have the sand packed uniformly. Any variation would produce a test bar of irregular section. In filling a vertical mould with iron the free surface is so small that bubbles of gas cannot reach the surface, but are caught near the walls of the mould.

A square bar cast flat fills with iron so slowly that all bubbles of gas and all impurities that would form spongy spots have ample time to rise to the surface, which will put all flaws in the top surface, where they will not weaken the test bar.

The top of the test bar should be marked, so that it will always be placed in the testing machine as it lay in the mould. In measuring such a square bar the depth can always be distinguished from the breadth, while this would be difficult with a round bar.

A horizontal mould can be rammed more uniformly than if vertical. The Western Foundrymen's Association appointed a large committee to investigate this subject and they reported that in one group all bars cast flat were perfect, while 43 per cent. of the round bars cast on end were defective. In another group 18 per

TABLE II.

			Silicon.	Sul.	Phos.	Mn.	Transverse.	Tensile.	Shrink.	Chill.
Furnace.	Heavy..	U.S. Eng. & Fd'y Co.	1.50%	....	....	....	......	......	....	......
		Lorain Fd'y Co....	1.20 to 1.40	.085	.24	37	3,900	37,962	....	......
	Medium..	Am. Fd'y Ass'n....	2.35	.044	.676	.43	3,200	29,000	....	......
Cupola.	Heavy....	Lorain Fd'y Co.....	1.20 to 1.50	.09	.35-.60	.50-.80	2,600	24,000	....	......
	Medium..	Lorain Fd'y Co.....	1.50 to 2.00	.08	.35-.60	.50-.80	2,400	23,000	....	......
	Heavy..	A. S. for T. M......	....	.12	....	....	2,533	24,000	....	......
		Am. Fd'y Ass'n....	1.50 to 2.00	....	....	....	......	......	....	......
		Loco-Cyl..........	1.25 to 1.75	.10	.90	....	2,070	......	....	......
		J. I. Case T. M. Co.	1.20 to 1.60	.095	.70	.70	2,400	22,000	.161″	.25″
	Medium	A. S. for T. M	....	.10	....	....	2,232	21,000	....	......
		Am. Fd'y Ass'n....	1.75 up.	....	....	....	......	......	....	......
		Pipe	....	....	....	....	2,227	......	....	......
		J. I. Case T. M. Co.	1.40 to 2.00	.085	.70	.70	2,200	20,000	.154″	.15″
		Phil. & Read. Ry...	1.40 to 2.00	.085	.60	.70	2,400	......	....	......
	Light...	A. S. for T. M......	....	.08	....	....	1,936	18,000	....	......
		Am. Fd'y Ass'n....	Above 1.75	....	....	....	......	......	....	......
		Pipe	....	....	....	....	2,089	......	....	......
		J. I. Case T. M. Co.	2.20 to 2.80	.085	.70	.70	2,000	18,000	.141″	.05″
		Phil. & Read. Ry...	2.00 to 2.80	.085	.60	.70	2,000	......	....	......
	Chemical Work L. F. Co......		1.10 to 1.35	.07	.25	.40-.60	......	......	....	......
	Brake Shoes P. & R. Ry.......		2.00 to 2.50	.15	.70	.70	2,900	......	....	......
Chilling Iron A. F. A..............			Below 1.00	....	....	....	......	......	....	......

All specifications are reduced to terms of a test-bar 1 inch square × 12 inches betwee supports. Tensile tests are per square inch, but as the size cast is not recorded, possibly thes values should be modified.

cent. of the square bars cast flat and 54 per cent. of the round bars cast on end were defective.

The committee of the American Foundrymen's Association say that a round test bar is more difficult to make and test than a square bar.

Professor Woolson of Columbia University tested the following test bars of Series 18, Figs. 432 and 433. All bars were cast 1.13 inch diameter from the same ladle and were turned to 1.065 inch diameter, and were tested on an Emery machine.

Round Bars Cast Flat. Tensile Test.

Broke at	25,000 lbs.	small spongy spot.
" "	25,500 "	" " "
Average	25,250 "	= 28,345 lbs. per sq. in.

Round Bars Cast on End. Tensile Test.

Broke at	14,600 lbs.	Bad spongy spot.
" "	16,300 "	Bad blow-hole.
" "	17,400 "	Blow-hole half through.
" "	20,200 "	Small blow-hole.
" "	21,300 "	Small shot in surface.
" "	22,400 "	Solid.
" "	22,5(0 "	Slight spongy spot.
" "	23,100 "	Solid.*
" "	23,400 "	Slight spongy spot.
" "	23,500 "	Solid.
" "	24,000 "	Solid.*
Average	20,791 "	= 23,340 lbs. per sq. in.

All bars except * had a large number of small blow-holes in the turned surfaces. There was not a flaw in the fracture of any of the large number of square test bars cast flat in this Series 18.

The argument for a round bar is that the grain is more uniform, and that the corners of a square test bar take from the strength of the bar. The average of 38 bars each of American Society of Mechanical Engineers tests (19 series) gave transverse strength of bars of 1 inch area, square 2,361 pounds, round 2,107 pounds. The average of all bars 1 inch area of American Foundrymen's Association tests (*A* to *E*) gave square 2,688, round 2,136 pounds.

Referring to Fig. 452, we see that strength depends largely upon the amount of metal farthest from the neutral axis. The

fibre distance of *a* is .50 inch, of *b* .685 inch, but the small portion that was stretched most gave way. The fibre distance of *c* is .565 inch, but the small amount of metal at the lower corner makes *c* weaker than *a*.

The strongest portion of the test bar is its surface. If a hole were bored lengthways through a test bar, it would not greatly decrease the strength of a test bar.

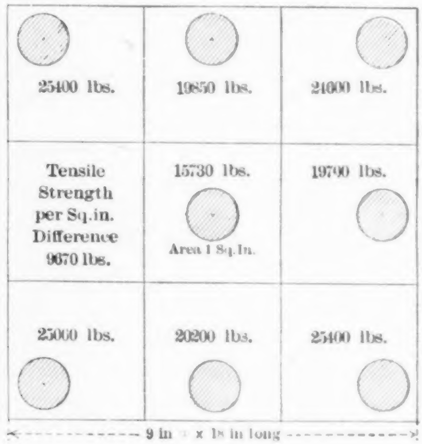


Fig. 455.

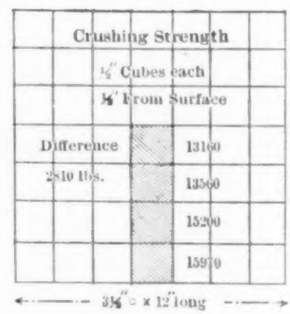


Fig. 456

The large square, Fig. 455, represents the end of a block of cast iron, 9 inches square and 18 inches long. T. C. 2.84, G. C. .60, Cd. C. 2.24, Silicon 1.10, P. .34, S. .09, Mn. .49.

This block was planed into nine parts, and from eight of these were turned test bars of 1 square inch area. The tensile strength of each is given in Fig. 455. The corner is 9,670 pounds stronger than the center, and the corner is 5,000 pounds stronger than the middle portion of the side.

This is also shown in Fig. 456 by compression tests of ½-inch cubes taken at each ½ inch in depth from a 3½-in square test bar. The centre of the side is 2,810 pounds stronger than the centre of the casting. It is upon this truss-like distribution of close grain that we depend for strength. See *e*, *f* and *h* of Fig. **452**.

For the greatest strength with the least metal we never make a cylindrical casting,

TABLE III.

Suggested Specifications Based upon Existing Practice.

Character of Casting.			Silicon range.	Sul. below.	Phos. below.	Mn. below.	Transverse 1 in. sq. × 12 in.	Tensile per sq. inch.	Shrink. 1 in. sq.	Chill 1 in. sq.	Ratio $\frac{\text{Tensile.}}{\text{Transverse.}}$
Furnace	Heavy		1.20 to 1.50	.085	.24	.37	3,900	38,000	......	.....	9.74
Furnace	Medium		1.50 to 2.00	.085	.68	.04	3,200	31,000	......	......	9.69
Cupola	Special	Heavy	1.20 to 1.50	.090	.60	.80	2,600	25,000	......	......	9.61
Cupola	Special	Medium	1.50 to 2.00	.080	.60	.80	2,400	23,000	......	......	9.58
Cupola	General	Heavy	1.20 to 1.75	.090	.70	.70	2,400	22,000	.161″	.25″	9.17
Cupola	General	Medium	1.40 to 2.00	.085	.70	.70	2,200	20,000	.154″	.15″	9.09
Cupola	General	Light	2.20 to 2.80	.085	.70	.70	2,000	18,000	.141″	.05″	9.00
Cupola	Chemical Work		1.10 to 1.35	.070	.25	.60	....	....	......	......	.....
Cupola	Brake Shoes		2.00 to 2.50	.150	.70	.70	2,900	28,000	......	......	9.69
Chilling Iron			Below 1.00.	......	......	......	....	....	......	......	.....

Transverse test-bars cast and tested 1 in. sq. × 12 inches long. Tensile test-bars cast 1.13 inch diameter and tested as cast.

Summary.

A variation of size of a casting causes a great variation in strength, because of the change in the rate of cooling.

A variation of shape of castings which have the same area of cross section causes a great variation in strength.

It is very difficult to calculate the strength of one form or size of test bar from the measured strength of another size.

A test bar should be cast horizontally in the ordinary way and in ordinary sand the same as other castings.

The average strength of at least two test bars cast together should be taken.

The distribution of metal in a square test bar gives a stronger casting than in a round bar of the same area of cross section, and more nearly represents the ordinary shape of castings.

A test bar 1 inch square is the size and shape in general use.

We think of transverse or tensile strength as so much per square inch.

DISCUSSION.

Mr. Thos. D. West.—In Mr. Keep's remarks on the utility of round bars, I note among other points which can be criticized his reference to the report of the Western Foundrymen's Association Committee on defective bars. I desire to say here

in the city where the tests were made, that that committee, which was a small working one, and not "large" as stated by Mr. Keep, used an appliance I sent them to make their round test bars. These bars were to furnish records of the strength, contraction, chill and fluidity of the iron, and in order to obtain all of these factors in one bar a design of flasks and patterns was used demanding a little of the moulder in the man who should make them. As Mr. Keep shows in his abstract of the report, the fact that quite a number of the flat bars were imperfect, is evidence that the casting work must have been done by those knowing little about moulding. This, I believe, was the case.

It seems strange that in Mr. Keep's researches to discover the utility and discuss the endorsement of round bars it should have escaped his notice that the American Foundrymen's Association Committee, which cast such bars in groups of one to two hundred in different shops at one pouring and made use of 1,229 test bars, ranging from half an inch to four and one-half inch in diameter, weighing about fifteen tons, found but about one-half dozen defective bars in the whole lot, and in some of the groups not a single defective bar was found, thereby demonstrating that round bars offer no difficulties in obtaining them solid. Pour round test bars from the top, which was not done with those cast by the Western Foundrymen's Association, and there will be no difficulty in obtaining perfect solid bars.

The Committee appointed by the American Foundrymen's Association, as well as that of the American Society for Testing Materials, have endorsed the adoption of the round bars cast on end as being preferable to a square bar. The latter is not condemned because of its reducing the strength of the bar, as stated by Mr. Keep, but because it is erratic on account of its corners being radically affected by variations in dampness of moulding sands and the temperature of metal when pouring. The smaller the bars the worse these evils.

*Mr. Edward J. Chambers.**—I entirely agree with the idea of learning all we can by the analysis of the pig iron that we use in the foundry and of testing it mechanically; but I cannot say that I am a convert to the use of round bars in preference to the rectangular sections. As you know, in England the rectangular section two-inch by one-inch bar has been a standard for testing for a long time, and when you come to think that cast iron is used

* Member of the Institution of Mechanical Engineers.

more in that form than it is in the round form, if you want the ordinary conditions represented in a test bar (namely, rigidity of the metal which is intended to be used mainly to carry a burden either as a girder or as a column), the rectangular section has a considerable advantage over the round. I notice that one of the conditions of the round bar is that it shall be cast on end. That strikes me as almost giving the show away, because if you are going to cast a round on end (I believe it has to be at least 15 or 16 inches long, as it is 12 inches between the centers of the supports), you are producing a totally different condition; you are really getting a condition of things in the test bar which does not exist in the castings. Added to which the remarks in reference to the strength of the different parts of the test bar, the corner as against the centers, it seems to me the rectangular bar carries out more nearly the conditions under which we make our castings. I should be very sorry to cast a slight upon theoretical work in reference to the foundry, but when you attempt to introduce an international specification you must be extremely careful that you do not commit the error which has been committed in the past by non-experts. I suppose when we look at the specifications which are put before us by some so-called professional men in reference to our mechanical work, that you will agree with me that they are often self-contradictory, and in many cases impossible to work to, and as soon as you, as a practical engineer, point out to the inspecting engineer the absurdity of some part, you will find, even if he does not tell you so, that it was put in by one of his juniors who got it from a text book, or else he tells you plainly that you may do what you think best. We as practical men must be very careful that we do not commit the error on the opposite side, and that is, make a specification that shall bar out a certain number of grades of pig iron which are most useful in manufacturing. I can say this from experience in wrought iron, and with regard to cast iron, the same argument would hold good, that there are many pig irons which by peculiar composition and mixture, will give a far better result in a casting than if you laid down a definite rule that each particular iron shall be of certain analysis. It is that point that I wanted to guard against. We might do a great deal of harm by barring certain irons. We must see that the iron is properly mixed; you must not leave the mixing of the iron entirely to the men.

I will say further that the testing of cast iron test bars is not to obtain resilience of the metal which has practically to be avoided in castings, but the stiffness of the metal which is the main purpose of all castings. A test bar 2 inch by 1 inch must of necessity be tested with the greater dimension vertical, as otherwise it would be no different in result to the 1-inch square bar per inch of width. The main reason for testing in this way is that in all castings for girders and similar work the metal is arranged relatively in this way, that is, of greater depth than width.

*Mr. L. W. Crosta.**—It seems to me the form of bar is not so well known in America as in England. If we are making special work, we are requested to cast bars with the same mixture of metals as the castings, and generally 2 inch by 1 inch by 3 feet 6 inches long, and which, when supported at 3 feet centers, must bear a weight of 27 to 32 cwts. at the center between the supports without breaking. With reference to torpedo and cylinder work, we always cast these "on end," and the test bar is required to be cast on the casting, which the inspector himself generally knocks off, and after being machined to his requirements, he takes, or sends it away for an independent test. The form of the bar does not trouble us in England.

Mr. Harrington Emerson.—About a year ago I was called to a large concern to advise them as to economies that might be effected in their shops. In looking over their foundry, which was under the management of a very skilful man, I told the owners frankly that it did not seem to me that I would be able to effect any improvement at all. Going, however, into the machine shop and taking time records of operations there it suddenly developed that we were experiencing very great losses owing to the variable quality of the iron that came from the foundry. We had records of the time in which certain machine operations could be carried out, and we found that we fell very much behind on these solely owing to the quality of the iron. We found that the average poor iron increased the expense very greatly. This foundry cast about ten tons a day, and we found we would be able to effect a saving of $18,000 a year, if the iron were uniformly good instead of variable as it was. In fact, the price of iron coming from the foundry was increased about $10 a ton owing to the extra labor that had to be put on it on account of its variable quality. It

* Member of the Institution of Mechanical Engineers.

therefore appeared that instead of being the department in which nothing was to be done the foundry was the place in which a very great deal could be saved. Not being a foundryman I began to look around to see where I could find help. I went to Detroit and hunted up Mr. Keep, and told him my troubles and asked him what I was to do. He very kindly gave me some of his tables, the result of his many studies, and I went back to the foundry, and, using the tables and methods that he had given me, we were able almost immediately to secure an absolutely uniform product of iron, not only then but regularly thereafter. I do not know what I would do in bettering foundry product if I could not turn to Mr. Keep's tables, and to the specifications that Mr. Keep has proposed, and I desire to take this occasion to thank him for what he has done to help those whose experience is so much less than his own.

Mr. W. J. Keep.—I may say that the American Waterworks Association has adopted a bar 2 inch by 1 inch deep and 2 feet long. I notice that the English bar is always three feet long, and is tested edgewise, the two inches being in the depth. I have wondered why that was done, and I have wondered whether the testing on edge was not rather to carry the impression that the iron was a very strong iron.

*President Wicksteed.**—Has it anything to do with the amount of deflection? With the elasticity of the material? Does it give you any more severe test as to the stretching properties and compressive properties of the material?

Mr. W. J. Keep.—I think not. I do not think it would be of any advantage. The only thing is that it gives you a very much higher strength per square inch.

---

* Member of the Institution of Mechanical Engineers.

No. 1042.*

THE POTENTIAL EFFICIENCY OF PRIME MOVERS.

BY C. V. KERR, NEW YORK, N. Y.
(Member of the Society.)

1. A water wheel is credited with an efficiency depending upon the proportion of the potential energy of the waterfall available that is converted into mechanical energy. Questions as to height of source or subsequent fall to sea level are not considered. A wheel located midway on the length of a river is charged with a certain volume of water per second falling from the level of the forebay to that of the tail race. The source of the river may be in the mountains hundreds of feet above the forebay, and the tail race may be many feet above sea level. That is the result of conditions imposed upon the engineer, and not the fault of the wheel.

2. Steam expanding adiabatically works at the expense of its own heat energy. From the initial condition as to pressure, moisture or superheat, expansion to the pressure of the exhaust renders available for conversion into work an amount of heat which is constant for each set of conditions. This amount fixes the limit to the economy of the engine. Whether the boilers could generate steam at higher pressure, with less moisture or more superheat and the piping system deliver the steam to the engine with smaller losses, or whether the condensing apparatus could maintain a higher vacuum are questions akin to the fall of water at the wheel, and are aside from the performance of the engine as a prime mover. It works between limits set by the designer of the power plant to meet existing conditions, and is not properly chargeable with what might have been done under different conditions.

3. The examination of a number of tests with this thought in

* Presented at the Chicago meeting (May and June, 1904) of the American Society of Mechanical Engineers, and forming part of Volume XXV. of the *Transactions*.

mind has brought out enough matter of interest to make it appear proper to present the subject for discussion. The term "potential efficiency" might be replaced by "kinetic efficiency" when a De Laval steam turbine or a Pelton wheel is under consideration, or it may even be thought that a new term is unwarranted; but, on the whole, the one presented has appeared admissible and most expressive.

I. Water Wheels.

4. The efficiency of a water wheel is easily expressed and easily comprehended. If the height of fall is H feet, and the flow in cubic feet per second is Q, the energy per second is $62.3\ Q\ H$; and if B is the brake horse-power developed, the work done per second is $550\ B$ foot pounds. Then the potential efficiency is

$$P = \frac{550\ B}{62.3\ QH} \quad \text{. . . . .} \quad (1)$$

5. In Table I are given the results of a number of tests on water wheels and the efficiencies are expressed in the usual way.

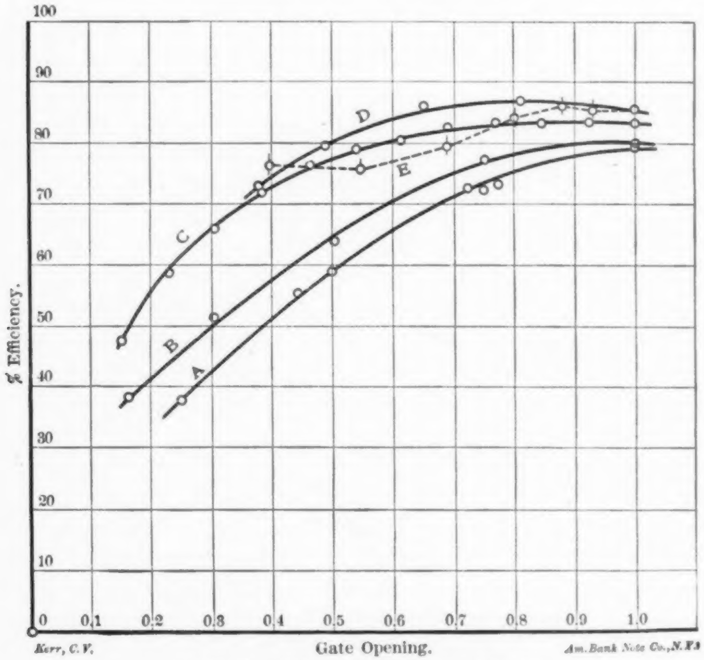
TABLE I.

POTENTIAL EFFICIENCY OF WATER-WHEELS.

NAME OF WHEEL.	Head in Feet. H.	Cubic Feet of Water per sec. Q.	R. P. M.	POWER. Available P. H. P.	POWER. Developed B. H. P.	Potential Efficiency in per cent. P.
Tremont	12.903	138.19	51.06	202.2	160.5	79.4
Booth	13.33	112.56	40.07	170.2	135.6	79.8
Boyden	16.6	147.1	63.5	277.0	222.04	80.2
Collins	16.59	113.46	63.38	213.1	131.49	85.1
Haenel	5.12	45.66	33.0	26.48	18.08	68.3
Tangential	570.84	6.84	210.	440.9	336.8	76.4
Swain	12.17	162.54	69.1	227.7	190.2	83.6
Hercules	16.96	88.33	140.62	169.7	145.72	85.8
Victor	11.65	45.86	144.5	60.52	52.54	86.8
Faesch & Picard	135.113	447.8	250.	6864.	5335. Electric	77.85
					5500. Brake	80.26
					611.	77.0
Pelton	658.	10.62	450.	793.	643.	81.0
					982.	83.1
Pelton	1919.	5.444	430.	1186.	1023.	86.2
Cascade	164.2	0.887	331.2	16.5	15.05	91.0
Centrifugal Pump	425.	2.47	890.	157.	118.9	76.0

Data for most of the wheels were obtained from Professor Wood's "Theory of Turbines," and some of the tests were made at the Holyoke testing flume. The data for the Pelton wheels were furnished by Mr. Henry, chief engineer for the Pelton Water Wheel Company. The Cascade wheel is a form of impulse wheel tested under the direction of Professor Hitchcock of the Ohio State University.

6. The results show the usefulness of the guide and vane type



Potential Efficiency of Water Wheels.

Fig. 457.

of wheel for low heads and the remarkable efficiency of the nozzle and bucket type under high heads. The test on the centrifugal pump is added to indicate the growing effectiveness of modern high-lift pumps.

7. In Fig. 457 curve *A* represents a series of tests with different gate openings on a Boyden outward flow turbine and on a Booth inward flow turbine. Curve *B* is for a Collins parallel flow wheel. Curves *C* and *D* are for inward and downward flow turbines, the

"Swain" and "Hercules" respectively. Curve E is for a Pelton wheel coupled to a 750 kilowatt generator, and operating under a nominal head of 1,960 feet, which is reduced by friction to about 1,920 feet in service. The speeds are 430 revolutions per minute and 170 feet per second at the rim. The regulation is by a deflecting needle nozzle, which accounts for the shape of the curve as compared with the others. The output was measured electrically, and the known efficiency of the generator was used to reduce results to brake horse-power for comparison.

II. Steam Engines.

8. The heat equivalent of a horse-power hour is 33000 × 60 ÷ 778 = 2545 B. T. U. Then, if H_1 is the initial total heat of the steam, H_2 the final total heat after adiabatic expansion, and W the weight of steam per horse-power hour by result of test, the potential efficiency is

$$P = \frac{2545}{W(H_1 - H_2)} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

9. If the steam is wet saturated, the initial total heat will be $H_1 = x_1 r_1 + q_1$ in which x_1 is the proportion of one pound vaporized, r_1 is the latent heat of evaporation and q_1 the heat of the liquid. Since the expansion is to be adiabatic no heat is transmitted to the expanding steam and the entropy remains constant. In symbols

$$\frac{x_1 r_1}{T_1} + \theta_1 = \frac{x_2 r_2}{T_2} + \theta_2 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

in which T_1 is the initial absolute temperature, or temperature Fahrenheit + 460, and θ_1 is the entropy of the liquid. A working idea of the nature of entropy may be gained from the term $\frac{x_1 r_1}{T_1}$ which expresses the quotient of the heat put into the steam during vaporization divided by the absolute temperature; and the term θ_1 is not esentially different, except that as the specific heat of water is not constant and the heat is put into the feed water at a changing temperature the numerical value of θ_1 is obtained by integration. Hence, for present purposes, entropy may be defined as the heat measured in British thermal units absorbed or

expended per degree absolute by the steam during a given change. Then from equation (3) the quality of the steam at exhaust will be

$$x_2 = \frac{T_2}{r_2}\left(\frac{x_1 r_1}{T_1} + \theta_1 - \theta_2\right) \quad . \quad . \quad . \quad . \quad . \quad (4)$$

and the final total heat,

$$H_2 = x_2 r_2 + q_2 = T_2\left(\frac{x_1 r_1}{T_1} + \theta - \theta_2\right) + q_2 \quad . \quad . \quad . \quad (5)$$

10. After finding in this way the values of H_1 and H_2 substitution in equation (2) determines the potential efficiency for given conditions. The symbols T_2, θ_2 and q_2, express the absolute temperature, entropy and heat of the liquid in exhaust. The values of entropy for steam and water are conveniently obtained from such books as Professor Peabody's "Tables of Saturated Steam" or Professor Reeve's "Thermodynamics of Heat Engines."

11. A word of caution may be timely here. On condensing work the vacuum is usually stated in inches of mercury whether measured by the gauge or by a mercury column. For the vacuum as given by a spring gauge a foreword of caution is proper—such gauges are seldom right. But if the vacuum is measured by a mercury column, as it should be, there still remains the question of the actual absolute pressure and temperature in the condenser.

If the mercury column reads 28 inches with the barometer at 29.8, the absolute pressure in the vacuum is 1.8 inches or $14.7\frac{1.8}{29.92} = 0.491 \times 1.8 = 0.885$ pounds per square inch. The steam tables show the corresponding temperature to be 98 degrees Fahr.; while, if the barometer stood at 30.4, the pressure in vacuum would be $0.491 \times 2.4 = 1.18$ pounds with a temperature of 107.6 degrees Fahr. This would make a serious difference in the amount of heat made available for conversion to work during adiabatic expansion to exhaust pressure.

12. If the steam is superheated, the initial total heat will be $H_1 = c_1(t_s - t_1) + r_1 + q_1 = c_1 s_1 + h_1$, where c_1 is the specific heat of superheated steam at the initial pressure, s the degrees of superheat and h_1 the initial total heat of dry saturated steam. And the entropy equation will be

$$c_1 \log_e \frac{T_s}{T_1} + \frac{r_1}{T_1} + \theta_1 = \frac{x^2 r^2}{T_2} + \theta_2 \quad . \quad . \quad . \quad . \quad . \quad (6)$$

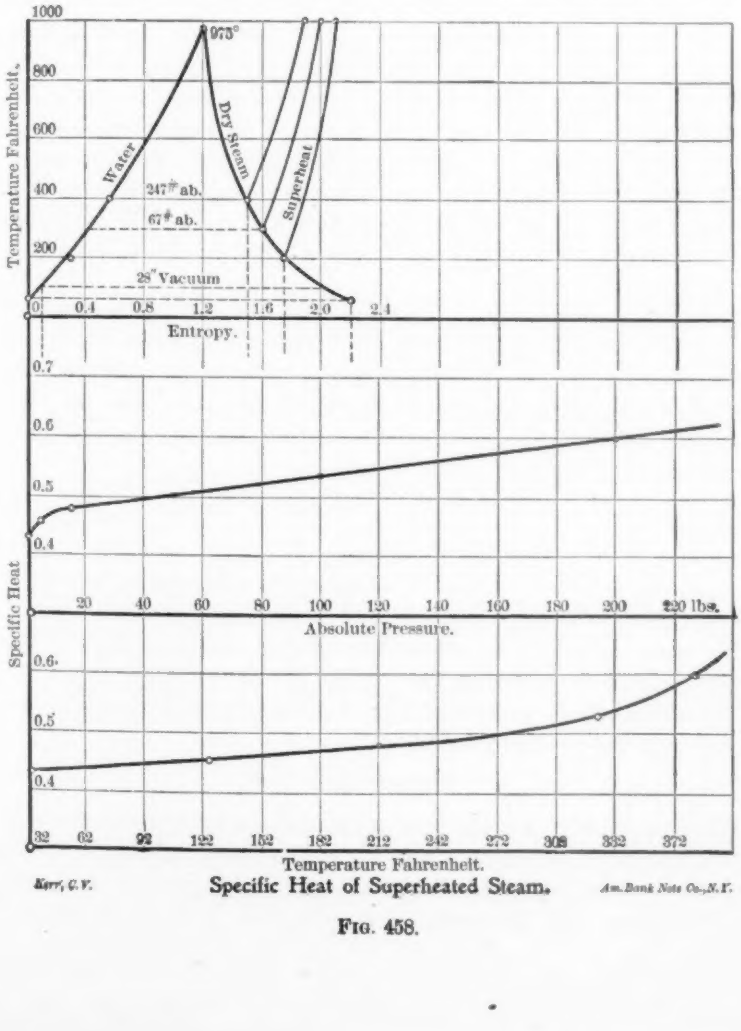
from which, $x_2 = \frac{T_2}{r_2}\left(c_1 \log_e \frac{T_s}{T_1} + \frac{r_1}{T} + \theta_1 - \theta_2\right)$. . . . . . (7)

and the final total heat,

$$H_2 = x_2 r + q_2 = T_2\left(c_1 \log_e \frac{T_s}{T_1} + \frac{r_1}{T_1} + \theta_1 - \theta_2\right) + q_2 \quad . \quad (8)$$

in which T_s is the absolute temperature of the steam, $\log_e$ is the hyperbolic logarithm and the other quantities are as before. The available heat is determined by $H_1 - H_2$.

13. Some uncertainty attaches to the value to be used for c the specific heat of steam at the various initial pressures and tem-



Specific Heat of Superheated Steam.

FIG. 458.

peratures. Enough has been done to make it certain that the value 0.48 established by Regnault for steam at atmospheric pressure does not apply to high pressures and superheats. Referring also to Fig. 458, in the entropy diagram, the curve for water approaches and finally meets the dry steam curve at the critical point for aqueous vapor. Accurate plotting on a large scale fixed this point at 975 degrees Fahr., which is quite near the theoretical value. Since the specific heat of water is known to increase with temperature, it may be inferred that the specific heat along the dry steam curve will also increase with temperature and finally equal the specific heat of water at the critical point.

14. On the basis of 0.48 at atmospheric pressure, some data furnished by Professor Jacobus for steam near 80 pounds gauge and the conclusion of Professor Bach that experiments should show a value close to 0.6, the curve of specific heat above absolute pressure in Fig. 458 was constructed, using the value 0.6 for 200 pounds absolute. Later researches, although inconclusive and somewhat discordant as to results, support the general correctness of this assumption. Since the function is a straight line above atmospheric pressure, it may be represented by the equation,

$$C = 0.48 + 0.00065\,p \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

Where p is the gauge pressure. Below atmospheric pressure, the function becomes a straight line if referred to temperature as a base with Hirns value of 0.43 at 32 degrees Fahr. and Regnault's 0.48 at 212 degrees Fahr. It may be represented by the equation,

$$C = 0.43 + 0.00028\,(t - 32) \quad . \quad . \quad . \quad . \quad . \quad (10)$$

where t is temperature Fahrenheit.

15. If the specific heat increases with the pressure and for any given pressure also as the amount of superheat, a conclusion apparently justified by researches already made, it may be suggested here that, within the region on the entropy-temperature diagram likely to be occupied by working steam, the specific heat may be represented by the equation,

$$C = a + bp + es \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (11)$$

Where $a = 0.48$, b is the increase in specific heat of dry saturated vapor at gauge pressure p, and e is the average increase in specific heat at constant pressure for an amount s of superheat in degrees.

16. After the foregoing discussion but little explanation is

thought to be needed for the tables and curves which follow. The computations necessary are applications of equation (2) to (10). The data used were obtained from engine-builders' catalogues, reports of tests published in engineering journals or directly from the parties concerned. In some instances where barometer as well as vacuum readings were not given, some injustice one way or the other may be done in comparing efficiencies.

TABLE II.

POTENTIAL EFFICIENCY OF STEAM-ENGINES.

Type.	Steam.			Vacuum Inches of Mercury.	Steam per I. H. P. Hour.	Available Heat.	Potential Efficiency.	Notes.
	Gauge Pressure.	Temperature of Saturated Steam.	Superheat at Engine.					
	Lbs.	*F.*	*F.*		*Lbs.*	*B. T. U.*	*P. C.*	
Double Vertical Single Acting........	100	338	0	0	26.19	147.8	65.8	20 × 16 Westinghouse Standard, 257 I. H. P. Shop test.
Vertical Single Acting Compound..	120	350	0	0 25	21.9 18.1	159.6 265.5	72.8 53.0	14 & 24 × 14 Westinghouse Compound, 170 I. H. P. Shop test.
Vertical Double Acting Compound ...	150	366	0	0	20.0	173.2	73.5	17 & 24 × 24 Westinghouse Compound. Rated 600 I. H. P. Shaft governor. Shop test.
Vertical Three Cylinder Compound.	184.6 185.6	381.5 382.0	0 0	25.19 27.25	12.24 11.93	286.9 315.3	72.5 67.6	43 5 & 2 – 73.5 × 60 Cylinders. I. H. P. 5,310. Economic rating 4,900–5,500 I. H. P. New York Edison-Waterside.
Horizontal Double Single Acting....	126 110	353 344	288 0	0 0	17.7 29.7	211.3 168.1	68.0 51.0	Trials "A" and "C," by Prof. Ewing, at Sheffield, England. Schmidt System.
Single Acting Tandem Compound..	158	369	219	26.14	11.75	361.0	60.0	Trial by Prof. Ewing, at Knocklong, Ireland. 65 I. H. P.
Double Acting Compound.	140	360	0 197	27.5 27	17.2 10.4	313.6 344.1	47.0 71.0	Trial by Prof. Ewing, at Middlepolder, Am. 125-184 I. H. P.
Single Acting Twin Tandem Comp'd.	157.3	369	295	26.87	8.97	370.1	76.7	Trial by Prof. Lewicki on Thale Engine. 257.6 I. H. P.
Double Acting Twin Tandem Comp'd.	140	360	331	25.4	8.96	352.9	80.5	Official Trial. 1,045 I. H. P. Schmidt Engine, Pabianice, Poland.
Four Cyl'der Triple Expansion Sulzer Engine............	184 188	381 382.7	0 223	26.25 28	11.57 8.97	305.7 392.1	72.0 72.3	3,000 I. H. P. Rated. See *Eng. News*, Oct. 2, 1902.
Horizontal Tandem Compound.......	132.2 128.9 131.6	356.6 354.8 356.2	0 98 310	27.9 27.8 27.8	12.06 11.00 8.86	315.7 330.6 384.7	67.0 70.0 74.7	Van den Kerchove Engine. Tested by Prof. Schröter. 220 I. H. P.
Vertical Triple Expansion Pumping Engine, 780 I. H.P.	155	368	0	26	11.38	279.0	80.2	20 M. Gal. Snow Pump. Test by Prof. Goss, at Indianapolis Water-Works.
Cross Compound Corliss............	145.1 142.4	363.2 361.8	0 374.5	25.24 26.79	13.84 9.56	279.2 387.1	65.8 68.8	Rice & Sargent Engine. Tested by Prof. Jacobus, at Millbourne Mills, Phila.
Cross Compound Corliss............	151.3	366.2	0	28.63	12.1	338.0	62.2	Rice & Sargent Engine. Tested by Prof. Jacobus for Amer. Sugar Ref. Co., Brooklyn.

17. From Table II it will be seen (1) that simple engines with a relatively high-water rate may stand well when compared on the basis of the proportion of available heat converted into work;

(2) that engines built to use superheated steam increase in potential efficiency with the amount of superheat.

18. The effect of size on economy is very well shown by the curve in Fig. 459. The two curves in Fig. 460 make it appear that an engine built to run either condensing or non-condensing may

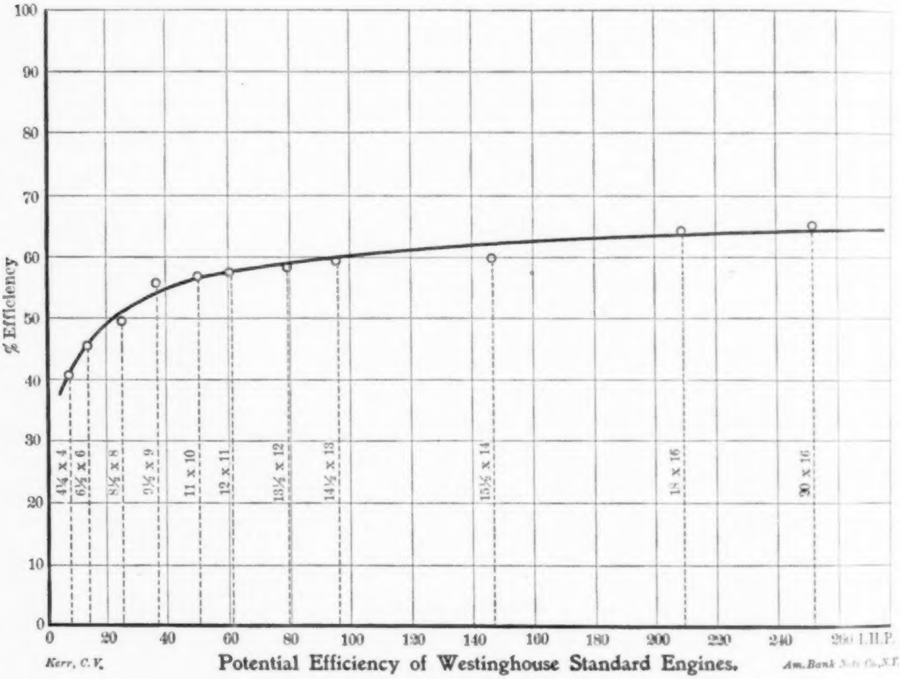
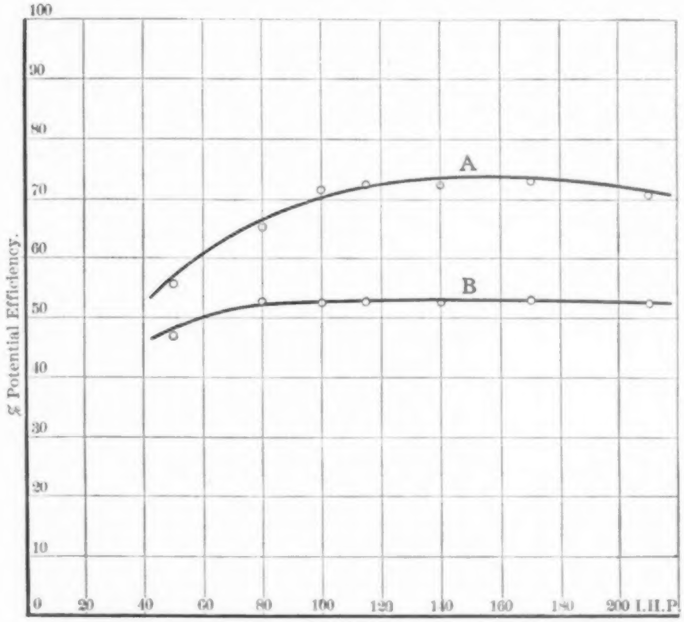


FIG. 459.

have not only a higher water rate, but also a higher potential efficiency on non-condensing work.

19. The lower curves in Fig. 461 show that for an engine rated at 5,200 indicated horse-power and running under the same conditions of steam pressure, load and vacuum a reheater is a little worse than useless. The upper curve passed through groups of tests with similar loads and pressures shows increasing efficiency for the higher back pressure. It also presents a means of estimating the water rate for conditions other than those of the tests. Thus in this case the contract called for a water rate of 12.5 pounds per indicated horse-power with rated load, dry steam at



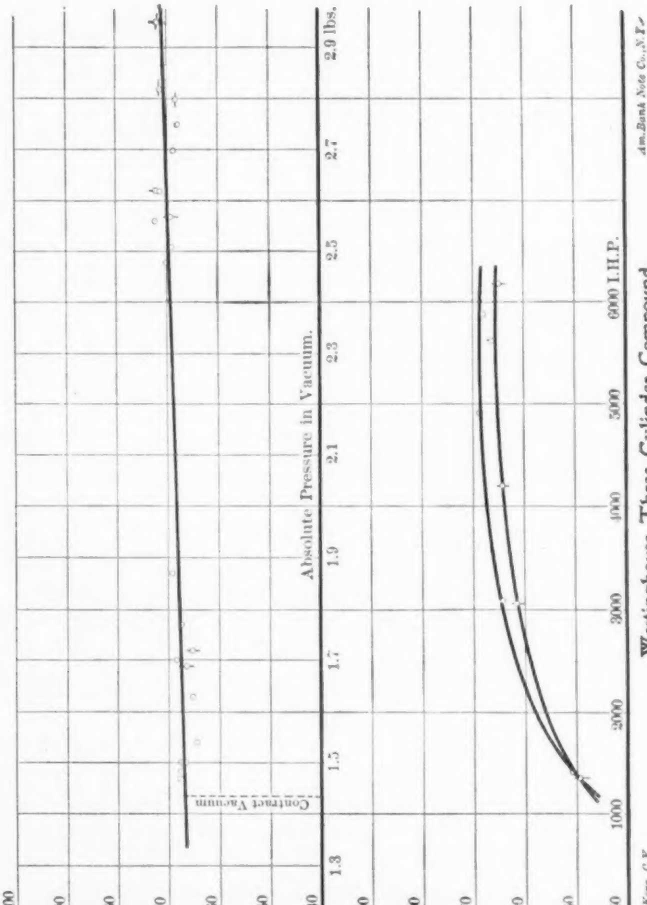
Potential Efficiency of Westinghouse 14" & 24" x 14" Compound.
A. Non-Condensing. B. Condensing.
120# Gauge, 25" Vacuum.

FIG. 480.

175 pounds gauge and a vacuum of 27 inches referred to normal atmospheric pressure. The available heat under adiabatic expansion is 313.2 British thermal units per pound of steam. Hence the water rate at the contract vacuum where the potential efficiency is 0.66 should be $W = 2545 \div 313.2 \times 0.66 = 12.3$ pounds

20. The curves *B*, Fig. 462, give the performance of a 16 & 30 × 42 cross compound Corliss condensing engine running at 120 revolutions per minute between 120 pounds gauge and 26-inch vacuum. Although rated at 400 indicated horse-power, it was developing only 100 horse-power. In comparison with this is the pair of curves *C* for the Van der Kerchove engines tested by Professor Schröter under full load. The steam pressure was 131 pounds gauge and the vacuum 27.8 inches. Both pairs of curves are practically straight lines, and show a continuous increase in efficiency with the amount of superheat.

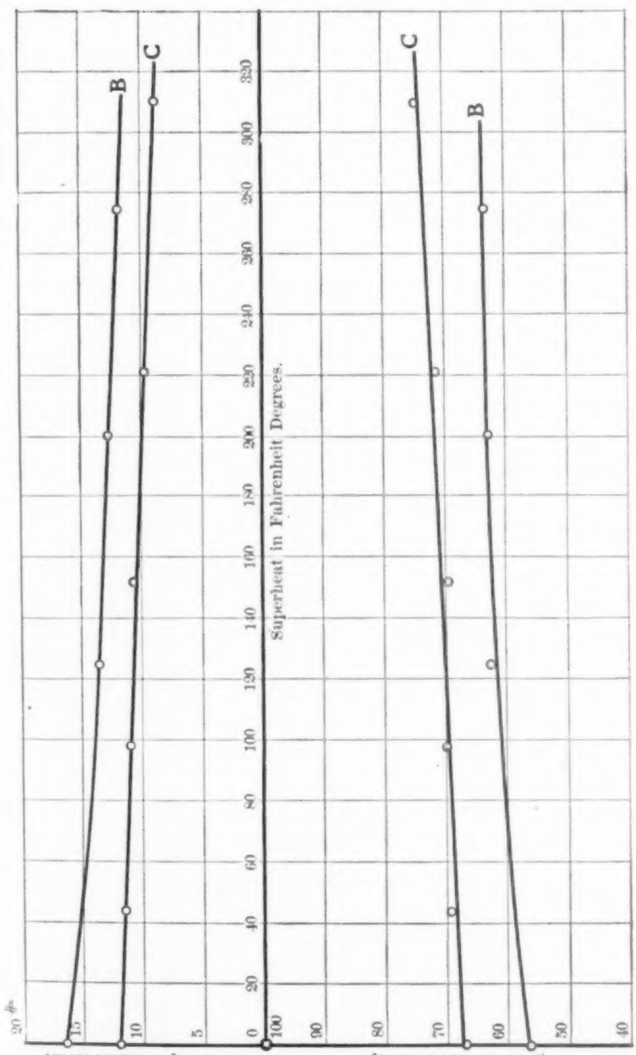
21. The Van der Kerchove engine was also tested with satu-



Westinghouse Three Cylinder Compound.
Waterside Station. New York.
Reheater. Jackets. Neither. Both.

FIG. 461.

rated and superheated steam under varying load, and the results are plotted in curves *A* and *B*, in Fig. 463. It will be noted that the curves are closely parallel, although the lower one is slightly the flatter. Curve *C* is for an engine of the same type to be designed and built for other conditions. Based on results obtained with the engines already built an efficiency of 63 per cent. on indicated horse-power is assumed for the new engine. The available heat is 352 British thermal units per pound, the mechanical efficiency of the engine is taken at 0.92 and of the generator at 0.97. Then the steam consumption per E. H. P. hour at best load will



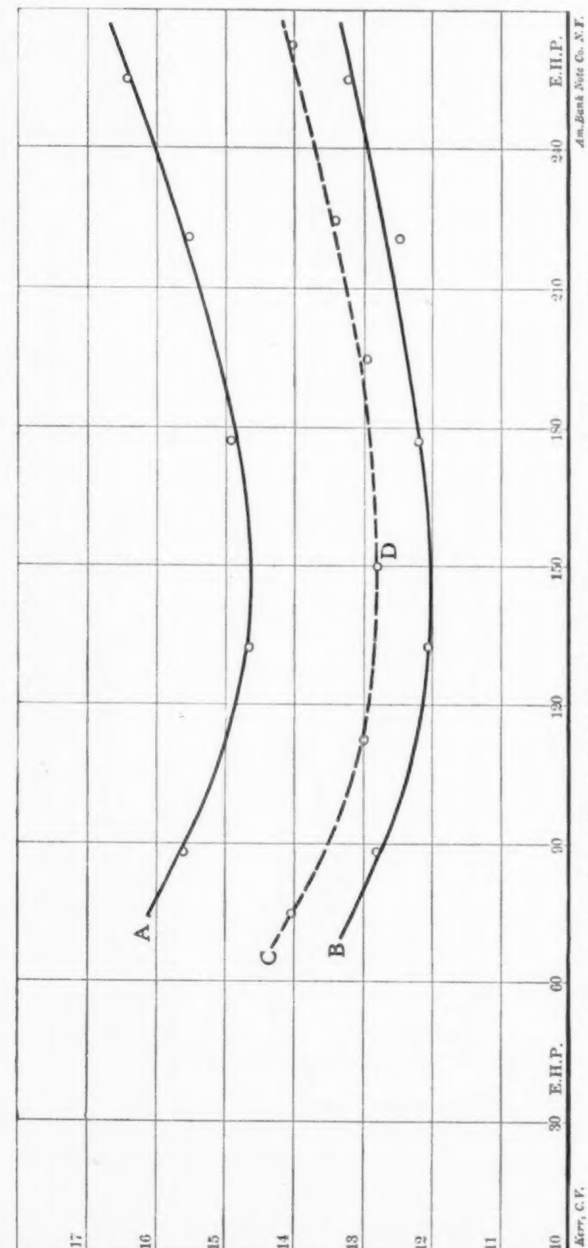
"B"—Corliss Cross Compound Condensing Engine.
"C"—Van der Kerchove Tandem Compound Condensing Engine.

FIG. 462.

be $W = 2{,}545 \div 352 \times 0.63 \times 0.92 \times 0.97 = 12.8$ pounds. This locates point D through which curve C is drawn parallel to A and B.

III. Steam Turbines.

22. This form of prime mover, nominally older than the Christian era, has in the last score of years taken on a very rapid



Van der Kerchove Engine, 131 lbs. Gauge, Saturated, 27.8 Vacuum, "A"
" " " " " " 225 F. Superheat " "B"
Proposed " 200 " " 50 F. " 28 " "C"

Fig. 463.

growth. The present size of units being built is a matter of common knowledge. The results of careful tests on sizes already in service are given in Table III. For the purpose of comparison with the performance of steam engines, the efficiency is based on the corresponding indicated horse-power. The mechanical efficiency is taken at 0.90 and the generator efficiency at 0.95, which combines at 85.5 per cent. This is not high enough for large vertical units, but probably quite correct for a general comparison.

TABLE III.

POTENTIAL EFFICIENCY OF STEAM TURBINES.

Type.	Steam. Gauge Pres.	Steam. Temp. Sat.	Steam. Superheat at Turbine	Vacuum inches of Mercury.	Water per H. P. Hour. Indicated.	Water per H. P. Hour. * Brake.	Water per H. P. Hour. * Electrical.	Available Heat.	Potential Efficiency.	Notes. Gen. Eff. = 95% Mech. Eff. = 90%.
	lbs.	F.	F.		lbs.	lbs.	lbs.	B. T. U	P. C.	
De La Val	206.2	390	0	26.6	13.26	14.73	15.5	318.0	60.4	Tests by Dean & Main on 200 K. W. Unit. Full Load.
	208.3	391	81	27.2	12.2	13.55	14.3	341.4	61.2	
Westinghouse 200 K. W. Unit	150	366	0	27	12.92		* 15.2	306.9	64.2	Full Load. Shop Test.
Westinghouse 400 K. W. Unit.	153	367	0	28	12.27	13.63	.....	325.0	63.9	Tests by Dean & Main. Full Load.
	151	366	182	28	10.13	11.25	.....	360.3	69.7	
Westinghouse 1,000 K. W. Unit.	149	365	0	28	12.6	.....	14.73	312.0	64.7	Full Load. Shop Test.
	136	358	56	28	11.2	.....	13.1	321.0	70.8	
	154	358	140	28	10.8	.....	12.66	341.0	69.2	
Westinghouse 1,250 K. W. Unit.	147.1	364.3	0	27.11	12.4	.....	14.52	304.1	67.5	Tests 6 & 11. Full Load. A. M Mattice, Power, March, '04.
	146.0	363.8	78.25	28.1	11.25	.....	13.17	340.3	66.5	
Westinghouse 1,500 K. W. Unit.	148	364.8	0	27	12.65	.....	14.8	301.7	66.8	Full Load Shop Test.
	146	363.7	28	27.5	11.68	.....	13.67	317.0	68.8	
Brown-Boveri Turbo-Alternator. 2,600 K. W. Unit.	173	376	196	27.75	9.84	.....	11.5	372.8	69.4 (+)	Frankfort. Corporation Test at Full Load. Exciter (+), or (−).
					9.5	.....	11.1		71.8 (−)	
Rateau Multicellular.	136.7	359	0	26.7	13.42	.....	15.7	293.4	64.7	525 E. H. P. Full Load. A. Rateau. Eng. Mag., Oct., '03.
Curtis 600 K. W.	140	360.7	0	28.5	12.2	.....	14.25	332.9	62.7	Full Load. W. L. R. Emmet. St.R.,Rev.4-20-03.
			150		10.68	.....	12.5	365.4	65.3	
Curtis 2,000 K. W.	156.0	368.6	212	28.5	9.76	.....	11.42	389.4	67.0	Eng'rs Club, Phila. 3-19-04.

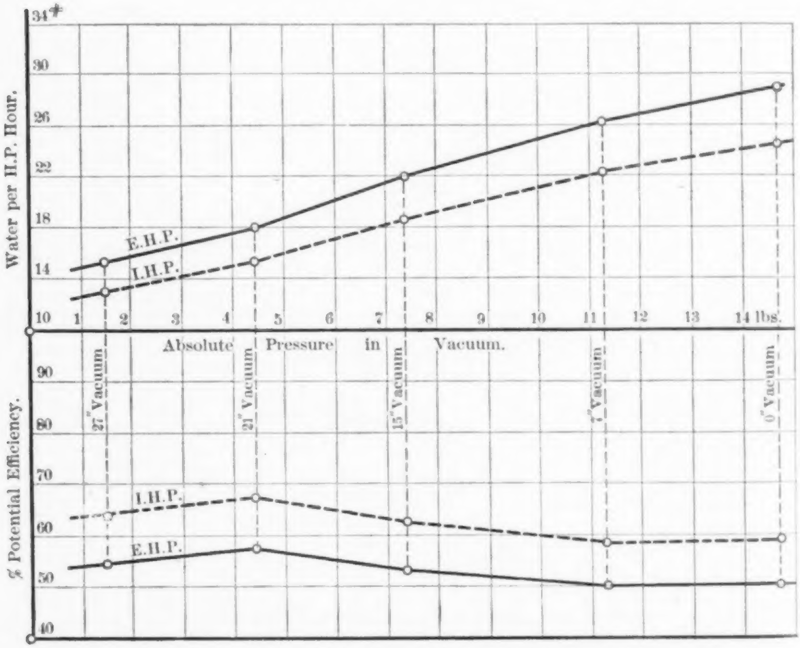
* Results of tests.

23. As in steam engines, the efficiency increases with the amount of superheat; but sufficient data are not at hand to permit the plotting of curves like those in Fig. 462. An interesting question is raised in the performance of a 200 kilowatt steam turbine at full load with constant steam pressure and varying vacuum. While the water rate decreases continuously the efficiency appears to pass a maximum at 21 inches vacuum, as shown by the curves in Fig. 464. A reason for this may be found in the rapidly increasing amount of heat available as the vacuum rises. This fact, together with the increasing cost of maintaining a high

vacuum, may be expected to locate some point which gives the minimum yearly total cost of power.

IV. Heat Efficiencies.

24. An extended discussion of all heat efficiencies proposed or current seems unnecessary. The Committee of the Institu-



Kerr, C. V. **Westinghouse Steam Turbine. 200 K.W. Full Load. Saturated Steam, 150 lbs. Gauge. Variable Vacuum** Am. Bank Note Co., N. Y.

FIG. 464.

tion of Civil Engineers defines "thermal efficiency" as the proportion of heat utilized to heat supplied; and they also define an "efficiency ratio" as the proportion which the thermal efficiency bears to the efficiency by the Rankine Cycle. In Volume XXIV., p. 755, *Transactions* American Society of Mechanical Engineers, the thermal efficiency ratio is the proportion which the heat equivalent of the power developed bears to the total amount of heat actually consumed as determined by test. In

a recent magazine article Professor Rateau explains the term "global" efficiency as the combined efficiency of turbine and dynamo as measured by the ratio of the actual consumption of the engine to the theoretical consumption of an assumed perfect engine. Doubtless a common idea underlies all this. The purpose of this paper is to give such an idea more definite form and a characteristic name.

DISCUSSION.

Mr. Sidney A. Reeve.—Mr. Kerr's paper is to be regarded as one of great interest and importance. The writer has frequently met, in past years, expressions from engineers of a desire for an efficiency percentage which should mean something. An efficiency percentage is generally understood to mean the portion of what a machine ought to do which it actually does do. This the thermodynamic efficiency plainly is not. To tell a man that the efficiency of a certain engine, for instance, is 7 per cent. imparts to him no information as to the worth of that engine unless there go with it a mass of other data which will enable him to figure out that the engine might possibly have attained a thermodynamic efficiency of, say, 14 per cent. From this secondary step, supposing that he carries with him the technical familiarity with such work to enable him to take it, he may draw the conclusion that the engine's performance is really one-half of what it might have been; or, in fact, that its true efficiency was 50 instead of 7 per cent.

It is this true statement of the engine's actual performance in proportion of what it might have done which Mr. Kerr calls its potential efficiency. For his excellent work in seeking the introduction of the habit of using this value as an indicator of the worth revealed by every engine-test we all of us owe him cordial gratitude. With the name with which he has christened it, however, I wish to quarrel slightly.

In the first place, while the energy available for work is of potential form in the water-wheel analogy from which he draws his illustration of the term, in the steam-engine it is not. Speaking more accurately, not all of it is potential in form. Much of it is kinetic thermal energy. Therefore, there is no active reason why the term "potential" should be used at all in this case—especially as he is forced to abandon it so soon as the steam-engine under consideration becomes turbine in form.

On the other hand, if this term which Mr. Kerr advocates be the true efficiency of any concrete thing, it is that of the engine-cylinder and its valves and ports. It expresses truly the skill with which the entire portion of the engine which is devoted to thermodynamic activities, as contrasted with that portion which is devoted to the transmission of the power developed on the piston to the shaft, performs its allotted task. It tells instantly the proportion of the possibilities theoretically open to the designer which he was able to develop, in so far as his skill was applied to the design of the cylinder and its adjuncts.

For this reason I have always called this efficiency the "cylinder-efficiency," and under that name it has been presented to my students, for the past eight years, as the prime indicator of the worth or the fault of an engine. It was given prominence in my text-book, "The Thermodynamics of Heat Engines," published some eighteen months ago. It was under this name that I urged its adoption, not only as one of the many efficiencies which might be assigned to an engine, but as the one of chief significance, upon the Committee of the Society which reported upon the proper system of engine-testing at the May meeting in Boston, in 1902. At that meeting I advocated the same step orally. But my letter never received either acknowledgement or reply, and my oral suggestion was unanimously approved by just one man. So I have been forced to feel that as a missionary I am a failure. I know how it must feel to have been eaten up. From the darkness and oblivion of the interior comes this voice handing over the honors and the tribulations of the task to Mr. Kerr.

In regard to Mr. Kerr's Table II., on page 918, of values evidenced by standard existing engines of varied types, I would raise some suspicion as to the values computed and reported there for the efficiency in question. This suspicion arises from the fact that many of the values are higher than any that I have ever found. In those engines using superheated steam I explain it as partly due to an erroneous assumption as to the specific heat of superheated steam. It having appeared plain to me some years ago that Regnault's value, 0.48, did not at all coincide with later investigations, I have gathered what recent observations have come to my hand in a paper upon the topic which is at present about to be published in the Journal of the Worcester Polytechnic Institute. In this search I found reason for recomputing the observations made by the late Mr. Grindley and

those of Dr. Griessmann at Dresden, by a method which seemed to me to be more accurate than theirs and which would bring the two into harmony upon a common basis which they did not before possess. The results of this method I present in the following table of total and specific heats of superheated steam.

It will be seen that the conditions never reach, as to pressure and temperature, those commonly occurring in engine-practice. As to what is the specific heat under those conditions I know nothing exact. From the field which is covered, however, it would appear plain that the specific heat is higher than the highest value stated by Mr. Kerr, 0.6. Using the assumed value of 0.75, therefore, which I consider none too high, I have recomputed a few of the values for the potential efficiency given on page 918 of his papers. The great uncertainty as to what degree of vacuum is meant by so many inches of mercury renders the comparison inexact; but it will serve approximately. All condenser-observations should always be stated either in the absolute inches of mercury or in degrees Fahrenheit. The latter is easier to observe and is much more exact than any sort of a pressure-gauge except a mercury-column.

	Mr. Kerr's values.	My own.
Professor Ewing's Sheffield test	68.0	63.4
" " Knocklong test	60.00	60.2
" " Middlepolder test	71.0	69.7
" Lewicki's Thale test	76.7	72.5
Schmidt engine at Pabiance	80.5	76.0
Professor Goss' Indianapolis test	80.2	77.2
" Jacobus' Millbourne test	68.8	61.1

It is plain that if the specific heat be higher than we now think that it is, we are deceiving ourselves as to the excellence of our engines using superheated steam and also as to the apparently poor efficiency of our superheaters.

As to Mr. Kerr's equations on page 915, I would suggest that they give an unnecessarily complex idea of the very simple computation required to state this "potential" or "cylinder" efficiency. My steam-tables contain all of the entropy-values needed, already computed. The task, for saturated steam, then reduces to (1) to look up the boiler-pressure level in the table and note the total heat and the steam-entropy; (2) to look up the exhaust-pressure level and note the water-heat, the water-entropy and the temperature; (3) subtract the second heat from the first;

(4) subtract the second entropy from the first, multiply by the absolute temperature of exhaust, and deduct the result from (3); the result is the heat available for work. Of course when superheat is present one logarithmic computation must be added to this programme.

Mr. Samuel Webber.—Permit me to offer the following corrections to the nomenclature in the Table of "Potential Efficeencies," on page 921—viz., there is no such thing as a Tremont turbine. When the "Boyden" turbines were being introduced into the mills at Lowell, the first one was put in at the "Appleton Mills." The success of this one led to their adoption by the other mills, and the one at the "Tremont Mills" was tested by Mr. Francis, who published his results as those of "the Tremont Turbine," but it was simply one of the Boyden wheels then being constructed for the different mills and not a distinct wheel.

The turbine at the "Booth Mills," which he also reported, was an experimental one of his own design, reversing the position of the guides and buckets from those of the Boyden or Fourneyson wheel, and was never duplicated. This method of "inward flow" was taken up by many other inventors, Swain, Leffel and others, and became the leading type since followed. The next step was to discard the inward discharge, and make that "downward," while the water entered from the circumference and made a "quarter turn." One of the earliest wheels of this type was the "Vandewater," which was improved by Theodore Risdon of Mount Holly, N. J., and one of which I tested at the Centennial Exposition in 1876, obtaining over 86 per cent. efficiency. This was soon followed by the "Victor," "Hercules" and McCormick wheels, all possessing the same general features, of fewer and larger buckets, lessening the peripheral friction, and admitting much larger volumes of water to be used in the same diameter of wheel. The results as collected by Mr. Kerr agree with my own tests and observations, and show that the late Prof. De Volson Wood was substantially correct in the conclusions at which he arrived in a paper published a few years since, that after making the necessary deductions for friction and "slip," about 85 per cent. was all the potential efficiency that could be expected from a turbine. The results of 90 per cent. or thereabouts, sometimes reported, have never, in my knowledge, been duplicated on second tests of the same wheel.

Mr. Francis Hodgkinson.—The method of comparing engine

performances proposed by Mr. C. V. Kerr is certainly the best method of making such comparison. There is, however, nothing new in this method. Professor Rateau makes use of it in showing his turbine efficiencies. I have always done so in prophesying the economy of Westinghouse-Parsons steam turbines.

The principal object of Mr. Kerr's paper seems to be to propose a new name for this efficiency; viz., "Potential Efficiency."

I am not sure that I think this a good name, because it is new and gives no clue as to what is meant.

I have found myself generally understood by using the term "Efficiency of the Rankin Cycle Efficiency." This is cumbersome, but I see no objection to the old-fashioned expression "Kinetic Efficiency" for this purpose.

Prof. D. S. Jacobus.—In the "Report of the Civil Engineers of London on Standards of Thermal Efficiency for Steam Engines" it is recommended that the ratio of the efficiency of an engine to that of an ideal engine working with the Rankine cycle be determined. This is also recommended in the "Report of the Committee of this Society on Standardizing Engine Tests." I would like to ask Mr. Kerr whether this ratio is not the same as the one which he gives.

Professor Reeve says that he made some suggestions to the committee in his discussion on Standardizing Engine Tests, and that this was the last he heard of his suggestions. Perhaps he did not read the Committee's final report because his ideas were followed to the extent of incorporating the factor for ratios of efficiency recommended by the Civil Engineers of London. Furthermore a chart is reproduced in the report from which by simple observation the heat consumption of the ideal engine under any set of conditions can be readily obtained.

Mr. Henry L. Doherty.—I regret to see a tendency to change from our old efficiency of rating. I am afraid that would tie us to a steam turbine or a steam engine, and we need a rating that applies to all sorts of prime movers. The internal combustion engine I believe would soon be on a basis where, considering the available heat that it might use, it would have an efficiency of over 100 per cent. You will have to have one basis for that and another basis for the steam engine.

Mr. C. W. Rice.—In reading Mr. Kerr's paper I notice that he mentions the recent test of an engine having a combined efficiency of 95 per cent. I would like to know from him whether

he accepts that. From his paper the theoretical possible efficiency is 98.24.

*Mr. C. V. Kerr.**—Replying to Mr. C. W. Rice, I would say that I most assuredly accept that value, as I helped to make the test.

It is gratifying to note the general agreement that the idea here called "potential efficiency" forms the best basis for comparing engine performances. What the technical name shall be is of less importance, but the one proposed is still believed to be peculiarly fitting. The numerical values of the "efficiency ratio" of the British Civil Engineers, of the "cylinder efficiency" of Professor Reeve, of the "global efficiency" of Professor Rateau, or of the "potential efficiency" as here defined may, when applied to the same conditions, be identical. But the term "potential efficiency" has the advantages of expressing what it is *possible* for the prime mover to do, of being directly derived as the "efficiency ratio," for instance is not, and of making direct and right comparisons between the performances of water wheels, steam engines and steam turbines.

For Professor Reeve's frank and earnest support of the idea involved and for his kind words in appreciation of my efforts I am grateful. And, if he or any one else will suggest a better term than "potential efficiency" which all will accept, let us have it; *but let the idea prevail.*

The differences in values of potential efficiency obtained by Professor Reeve in using 0.75 as the specific heat of superheated steam, as compared with my own results are arguments for researches establishing the facts of superheated steam, as Regnault did for saturated steam, rather than against the common idea. My paper purposely gives in full the data and methods used, so that in event of the facts being established the proper corrections can be made in future comparisons.

Most engineers who follow such methods intelligently will soon discover the short cuts in calculation of results that are permissible. The use of charts such as that to which Professor Jacobus calls attention have their advantages. Of greater convenience in exact work under test conditions are the entropy values in Professor Reeve's steam tables.

Mr. Doherty's expression of regret arouses my respect for his veneration of the aged. But the farmer finds the self-binder better

* **Author's closure, under the Rules.**

for his harvest than the cradle, and the housewife prefers the power loom in the factory rather than the spinning-wheel in the house. The direct driving of generators by water wheels, steam turbines and engines, only one of these prime movers yielding an "indicator card," makes a modern method of comparing efficiencies desirable.

A clear understanding of the meaning of potential efficiency would probably have saved Mr. Doherty from the error of supposing that the efficiency of the internal combustion engine on the basis of available heat could be over 100 per cent. The potential efficiency of the other prime movers named really depends upon the potential energy available in a pound of water used in the natural state or as steam. If the specific heat of gases did not change with high temperature, or if the law of change were known, we could compute the initial temperature and pressure possible in the gas engine cylinder, and assuming adiabatic expansion, compute the energy available per pound of explosive mixture. For the further reason that so few gas engine tests are made or reported with the data here needed, I decided to omit a comparison of gas engine performance with that of other prime movers. This merely delays a complete comparison of all prime movers on the same basis, and is no excuse for not using the data or methods at hand.

The historical matter furnished by Mr. Webber is of interest and has led to slight changes in the text. His support of the correctness of results verifies the conclusion based on Tables I, II and III, that the steam engine or turbine has not reached the mark of potential efficiency set by the water wheels.

The output of a generator is conveniently measured in terms of the kilowatt hour, the heat equivalent of which is 3,412 British thermal units. If the generator is driven by a steam turbine or engine, the potential efficiency is expressed by

$$P = \frac{3{,}412}{w\,(H_1 - H_2)}$$

in which $3{,}412 = 2{,}545 \times 1.34$ and w is the observed water rate per kilowatt hour. Hence the change in heat equivalent does not cover the mechanical efficiency of prime mover and generator. If the mechanical efficiency is about 0.90 and the 3,412 be changed to 3,800, then

$$P = \frac{2{,}545}{w_1\,(H_1 - H_2)} = \frac{3{,}800}{w_2\,(H_1 - H_2)}$$

in which w' is the observed water rate per indicated horse-power, and w is the water rate per kilowatt hour. Or, if the mechanical efficiency is about 0.85, the 3,412 becomes 4,000 and

$$P = \frac{4{,}000}{w\,(H_1 - H_2)}$$

In the case of the water wheel driving the generator, the potential efficiency will be

$$P = \frac{737.3}{62.3\,QH}$$

since the mechanical equivalent of a kilowatt is 737.3 foot pounds per second.

To compare the performance of prime movers driving generators, it will be convenient to make

$$P = \frac{3{,}412}{w\,(H_1 - H_2)} \text{ or } \frac{737.3}{62.3\,QH}$$

which may be called the commercial or practical potential efficiency.

No. 1043.*

MIDDLESBOROUGH DOCK ELECTRIC AND HYDRAULIC POWER PLANT.

BY VINCENT L. RAVEN, CHIEF ASSISTANT MECHANICAL ENGINEER, NORTH EASTERN RAILWAY.

(Member Institution Mechanical Engineers.)

1. Now that the question of electrical appliances for dock purposes is so generally considered, the author thinks that it may be of interest to record and place before the members of the Institution the particulars of the installation and the tests which have been made at Middlesborough Dock, on the North Eastern Railway, comparing hydraulic with electric. These are working side by side, and are all thoroughly up to date, giving an excellent opportunity for judging the value of the one against the other, so far as economy in working is concerned. He proposes therefore first to describe the machinery at this dock, and then to give the very careful tests which have been carried out.

2. When the dock, Fig. 465, was constructed, it was equipped with steam cranes, and had a small hydraulic power station for working the bridge and gate machinery. Twelve years later, a further extension was made, and hydraulic cranes were then supplied to the new extension. A separate electric lighting station was also put down for dock lighting. With this plant, in the year 1900 the following tonnage was dealt with:—

	Tons.
Coal and coke exported	222,616
Merchandise, exported	197,933
" imported	36,941
Ballast worked	1,751
Or a total of	459,241

* Presented at the Chicago meeting (May and June, 1904) of the American Society of Mechanical Engineers, and forming part of Volume XXV. of the *Transactions*.

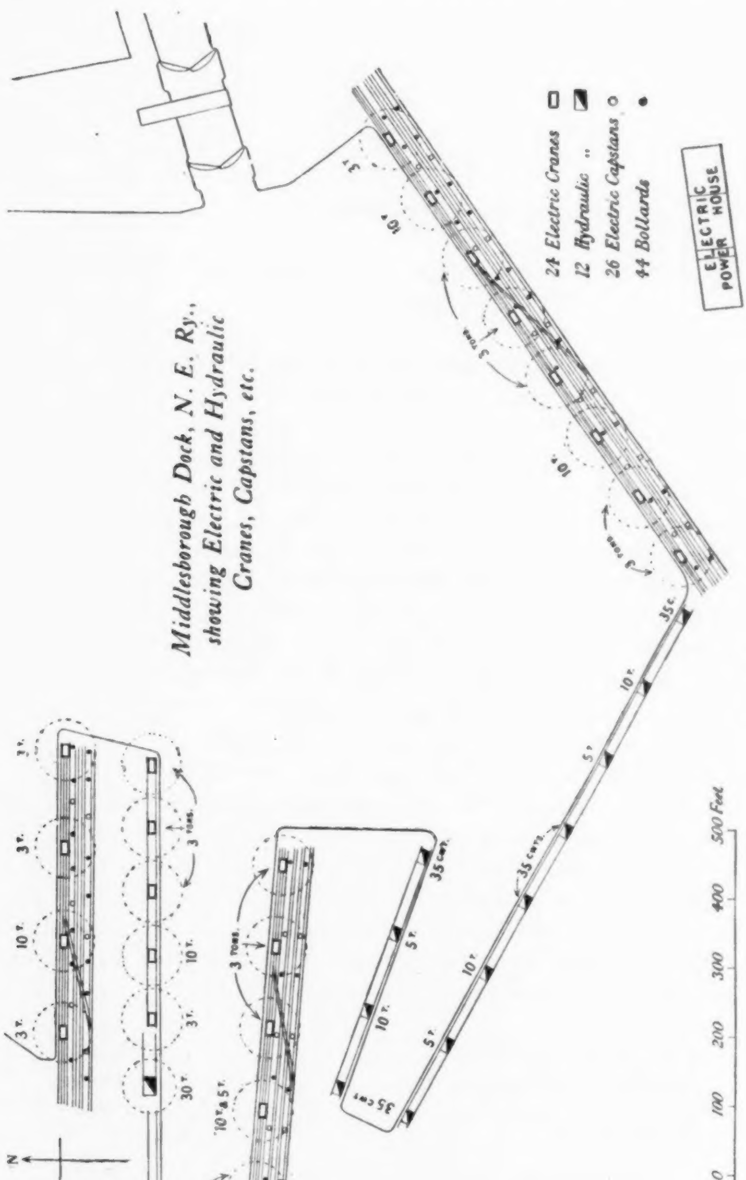

Middlesborough Dock, N. E. Ry., showing Electric and Hydraulic Cranes, Capstans, etc.
24 Electric Cranes
12 Hydraulic ..
26 Electric Capstans
44 Bollards
ELECTRIC POWER HOUSE
N
Feet 100 50 0 100 200 300 400 500 Feet

FIG. 465.

To do this work:—

	Coal. Tons.	Cwts.
The steam cranes burnt	1,989	0
The hydraulic station burnt	994	15
The electric station burnt	932	11
Or a total of	3,916	6

The total merchandise worked per ton of coal burnt was therefore 117 tons.

3. When the present extension was proposed, after careful consideration it was decided to abandon the steam cranes and to supply their places with electric cranes, and, at the same time, make one large power station, which would supply both the electric power for the cranes and for the lighting; and the hydraulic power, water for the hydraulic plant. The work in connection with the power station was divided into two contracts, Messrs. Siemens Brothers & Co., of Woolwich, being responsible for the electrical portion, and Messrs. Carrick & Wardale, of Gateshead-on-Tyne, for the hydraulic installation.

4. The power station contains three sets of Messrs. Belliss & Morcom's compound three-crank self-lubricating quick-revolution vertical engines, discharging with independent condensing plants, each developing 360 B.H.P. at a steam-pressure of 150 pounds per square inch at a speed of 380 revolutions per minute; these engines are direct coupled to three Siemens' dynamos, which have an output of 240 kilowatts at 430 volts when running at a speed of 380 revolutions per minute.

5. The engines on a six hours' test-run gave the following results:—

Steam pressure on range	154 lbs. per square inch.
Steam pressure in steam chest	128 lbs. per square inch.
Vacuum	24 inches.
Revolutions per minute	380.
Volts	430.
Ampères	552.5.
Exciting ampères	6.75.
Kilowatts	237.5.
Water consumption	21.1 lbs. per electric H. P.*
Efficiency	$\frac{318.3 \text{ E. H. P.}}{359.321 \text{ I. H. P.}\dagger} = 88.6$ per cent.

6. The switchboard is of marble, and consists of four machine and one feeder panel, in addition to the lighting panel. The in-

* Exclusive of steam used by the condensing plant.

† Exclusive of power abstracted by the condensing plant.

struments are of the moving coil type. The shunt winding of each dynamo is brought to a switch on the board, and these shunt switches are interlocked with their respective main switches, so that it is impossible to break the exciting circuit of any machine until the armature had been cut out of circuit, nor can the armature be thrown in until the shunt circuit is made. The three machines are run in parallel upon the bus bars, and the shunt circuits are excited from the bars.

7. The three hydraulic engines are of the triple-expansion surface-condensing inverted vertical marine type. The pressure pumps, three in number, are of the ordinary ram type, worked directly off the piston-rods, and each set is capable of delivering 300 gallons per minute at a maximum speed of 240 feet, against an accumulator pressure of 800 pounds per square inch. Both the hydraulic and electric engines can also be worked non-condensing, and the piping and valves are so arranged that either engine can be changed over from condensing to non-condensing, or *vice versa*, without any stoppage whatever. There are two hydraulic accumulators, 20 inches diameter by 23 feet stroke, and situated betwixt these is a large water-tank, which holds 12,000 gallons for supply to the pressure pumps.

8. The hydraulic engines on a six hours' test-run gave the following results:—

Water pumped at a piston speed of 240 feet per minute	311 gallons.
Water pumped per cwt. of coal burnt....................	5,090 "
Steam consumption per I. H. P. per hour..................	14.1 lbs.*
Total I. H. P..	181.5.†
Steam pressure per square inch...........................	150 lbs.
Vacuum..	28.5 inches.
Efficiency.............................. $\frac{\text{Pump H. P. } 162.5}{\text{I. H. P. } 181.2} =$	89.5 per cent.
Accumulator pressure.....................................	750 lbs. per sq. in.

9. There are two sets of condensing plant in connection with the electric steam-engines, each capable of dealing with 15,000 pounds of steam per hour; these were supplied by Messrs. Carrick & Wardale, of Gateshead-on-Tyne. Each set comprises a compound steam-driven air and circulating pump, together with a surface condenser combined on one bedplate. The circulating water is passed zigzag through the condenser, and the quantity used

* Exclusive of the steam used by the circulating pump.

† Exclusive of the power absorbed by the circulating pump.

is thirty times the amount of steam dealt with, and the vacuum maintained from 27 to 29 inches. The circulating water is drawn from the dock, a distance of 300 feet through an 18-inch suction-pipe, and at low water these pumps have a lift of 15 feet, but no difficulty has been experienced in getting the water even at such a great distance. There is also a circulating pump, fitted by the same makers, for supplying water to the condensers of the hydraulic engines, and it is similar in design to the electric plant. The three feed pumps and filters were supplied by Messrs. Henry Watson & Sons, High Bridge Works, Newcastle-on-Tyne.

10. The boilers for both plants are of the ordinary Lancashire type, six in number, 30 feet long, 8 feet 6 inches diameter, working pressure 160 pounds per square inch, and were made by Messrs. H. & T. Danks, of Netherton. The boilers are fitted with Proctors' shovel stokers, and the coal is supplied thereto by a coal elevator and conveyer supplied by Messrs. Graham Morton & Co., of Leeds, the coal in the first instance being raised from ground level in truck by means of a diagonal hydraulic hoist to the top of the coal cells and there discharged. A Green's Economizer of 384 tubes, 9 feet long, $4\frac{9}{16}$ inches diameter, is fitted in connection with the boilers.

The boilers on a six hours' test gave the following results:—

Grate area	38 square feet.
Coal burnt	5,208 lbs.
Clinker and ashes	356 lbs., or 7 per cent. of coal burnt.
Net consumption of combustible	4,852 lbs.
Combustible burnt per hour	809 lbs.
Combustible burnt per square foot of grate area per hour	21.3 lbs.
Pounds of water evaporated	36,300
Pounds of water evaporated per hour	6,050
Pounds of water evaporated per pound of coal burnt	7.5 lbs.

11. Feed water from hot well at 110 degrees Fahr. was run to waste on account of measuring the water, and town water was put in feed tank at a temperature of 54 degrees Fahr., or equal to a loss of 241 pounds of coal.

The steam and hydraulic mains are laid in circuit in the engine and boiler houses respectively.

12. The contract for the electric cranes and capstans consisted of nineteen 3-ton and five 10-ton cranes, Figs. 466 and 467, and twenty-six 1-ton pull capstans, Fig. 468. The whole of the elec-

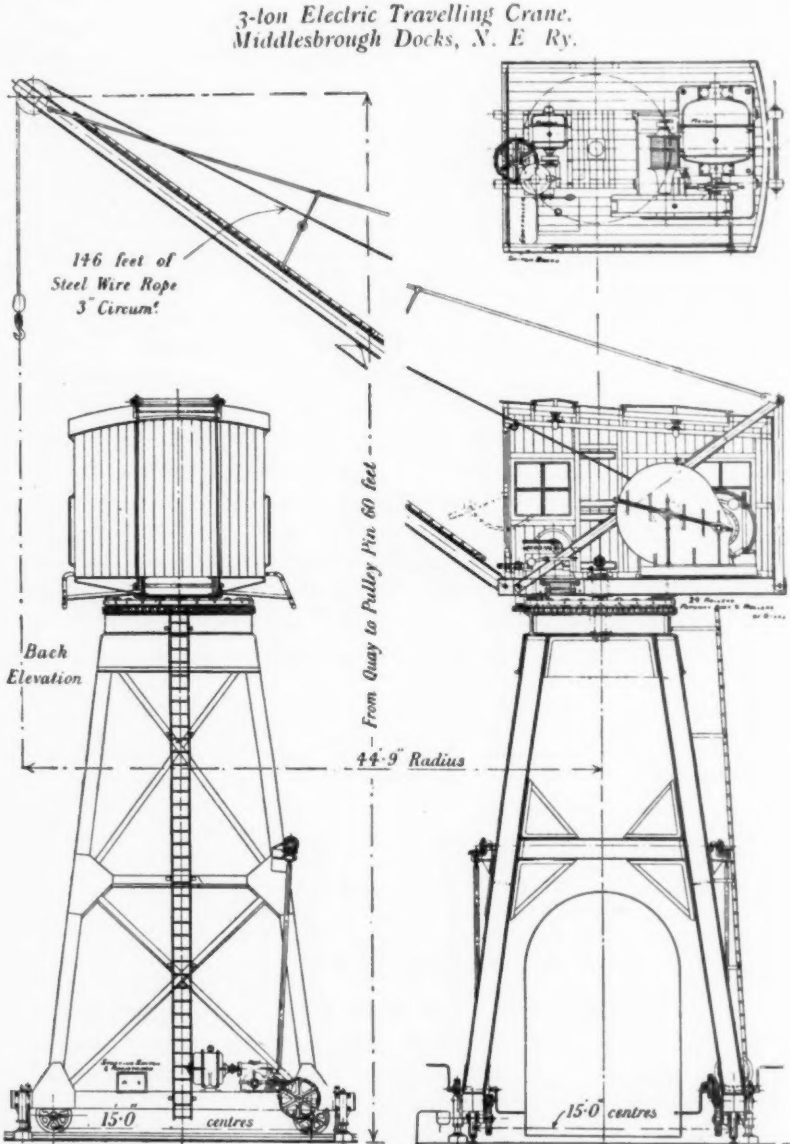

3-ton Electric Travelling Crane.
Middlesbrough Docks, N. E. Ry.
146 feet of
Steel Wire Rope
3" Circumf.
Back
Elevation
From Quay to Pulley Pin 60 feet
44'-9" Radius
15'-0" centres
15'-0" centres
0
5
10
15
20
25 Feet

FIG. 466.

10-ton Electric Travelling Crane.
Middlesbrough Docks, N. E. Ry.

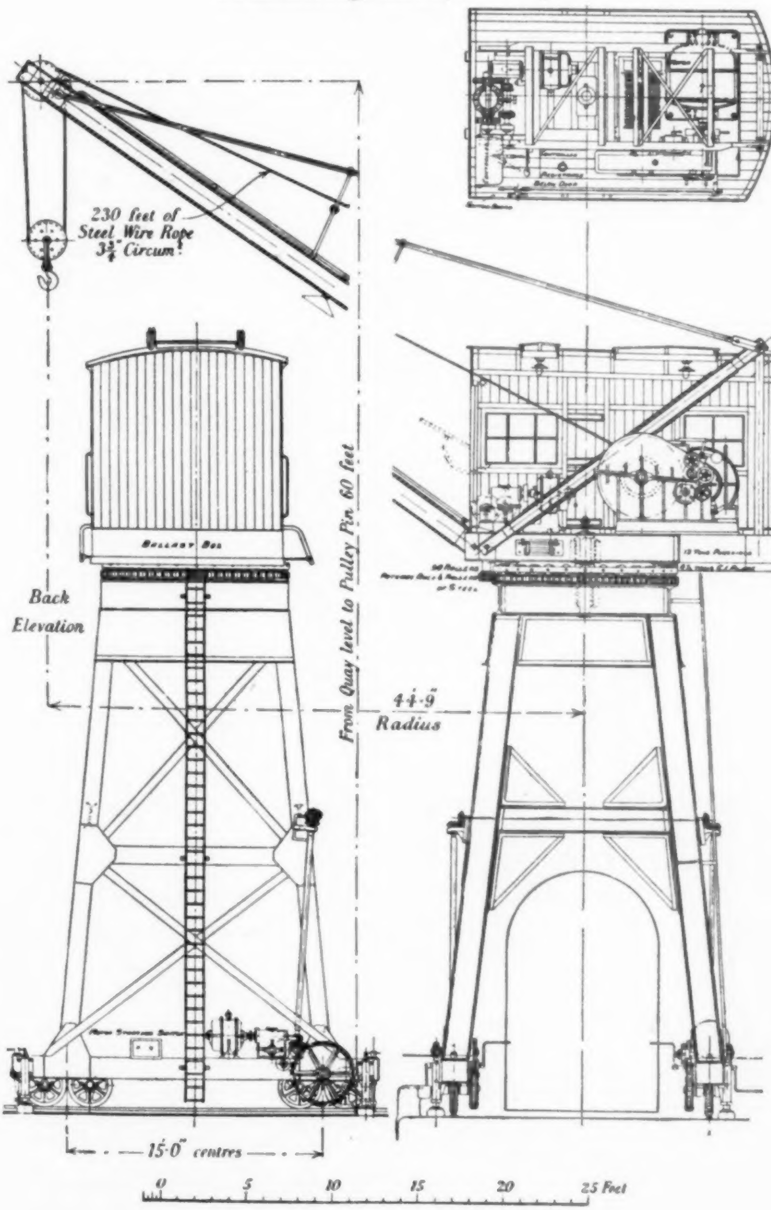


FIG. 467.

1-ton Electric Capstan.

Middlesbrough and Hartlepool Docks, N. E. Ry.

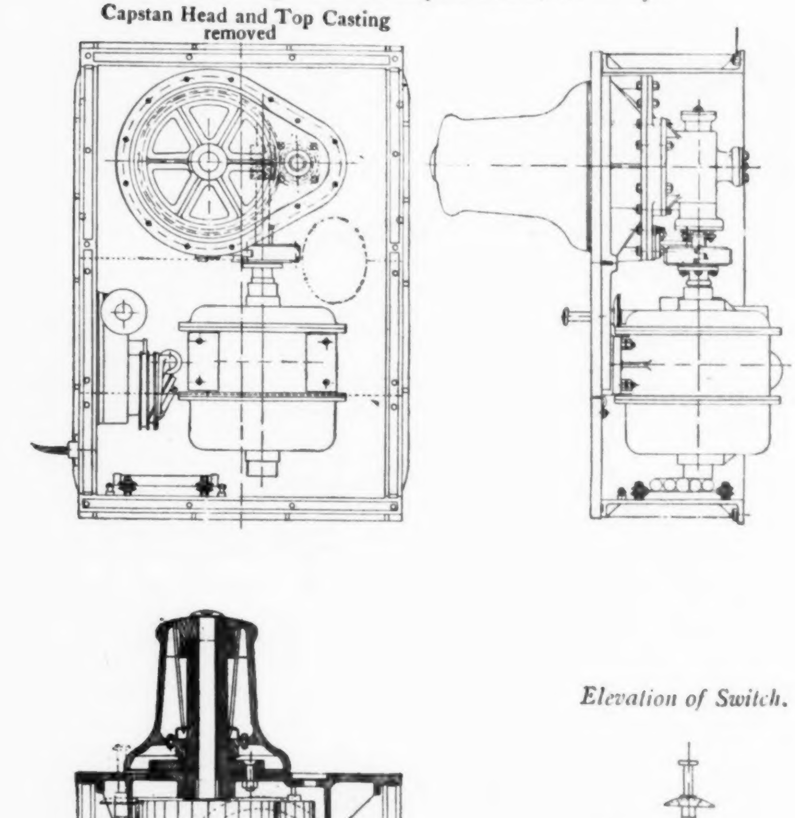


Fig. 468.

trical equipment for these was supplied under sub-contract by Messrs. Siemens Brothers & Co., of Woolwich.

13. The following are the principal particulars of the 3-ton cranes, Fig. 466, as specified:—

Height of jib top pin from quay level	60 feet.
Radius of jib	44 feet 9 inches.
Speed of lift, 3-ton load	150 feet per minute.
Speed of lift, 1½-ton load	250 feet per minute.
Revolving speed at hook	400 feet per minute.
Travelling speed	30 feet per minute.
Lifting motor, 50 brake horse-power	300 revs. per minute.
Revolving motor, 8 brake horse-power	1,000 revs. per minute.

14. The cranes are of the high gantry type, with gantry 30 feet high, and so constructed as to admit of the passage of wagons and engines underneath. The whole structure is rigidly tied together. Holding-down clips for fixing to the rails and screw-down jacks for increasing the base when the maximum load is being dealt with are also provided, but the cranes are absolutely stable, even with a 50 per cent. overload, without these attachments.

15. The travelling gear motion is fitted with both hand and electrical gear, the latter consisting of an enclosed series-wound multipolar motor, which is coupled to a steel worm working into a delta metal worm-wheel. The power is communicated to the trodwheels by means of spur and bevel gearing, the wheels on each side being coupled together by means of a cross shaft, thereby insuring uniformity of movement and avoiding any twisting strain on the gantry. The switch gear for the travelling motor is placed in a water-tight case upon the bottom framing of the gantry. To prevent the possibility of unauthorized persons starting the gear, the switch is provided with a detachable key, which is kept in the charge of the crane-driver. The top framing is constructed of H beams riveted up into box shape and carrying the heavy plate which takes the roller path. H beam cross-stays take the thrust and stiffen the structure. The roller path and rack are of cast-steel; they are not bolted to the top plate, and whilst they are held down by a groove turned out of the latter, they admit of some movement and ease the crane in case of a sudden stop when revolving. The rollers are twenty-four in number and of cast steel.

16. The collector is fitted to the center pillar, through which the mains are passed and attached to (and consists of) two copper rings mounted upon ambroin and separated by rings of the same

material. The collecting brushes consist of two gauze straps, each encircling one ring, the tension being regulated by springs. These straps are attached to terminals, which in turn are mounted upon porcelain insulators, from which the cables for the wiring of the cranes are taken. The crane cabin is of teak, and the front consists of a wrought-iron framing, glazed to enable the driver to have full view of his load throughout the operation. The cabin contains the lifting and sluing gear and the controlling apparatus.

17. The lifting gear consists of a barrel, wheel, and pinion. The wrought-steel pinion is cut from a blank, forged solid with the shaft, which is fitted with a half-coupling to meet that upon the motor spindle. Upon this shaft there is a band brake, actuated by an electro-magnet which will be described later. The pinion gears into a cast-iron spur-wheel keyed on the barrel shaft. Both pinion and wheel have machine-cut teeth, and work in a cast-iron gear case, which forms an oil-bath.

18. The revolving gear consists of a motor, worm, worm-wheel and spur-gear. The worm is cut from a blank, forged solid with the shaft, which in turn is fitted with a coupling to meet the motor. The worm engages with a delta metal worm-wheel, and a pinion keyed to the worm-wheel shaft engages with a spur wheel which is fitted to a short shaft actuating a pinion. This pinion travels round the rack of the crane and thus slues the cabin and jib. A brake actuated by a foot-treadle is fitted to the first-motion spindle, so that the sluing motion is always under control. The sluing motor is of similar type to the lifting.

19. The electric brake magnet, referred to previously, offers some points of interest. It is placed in series with the armature of the lifting motor, and consists of a horse-shoe magnet (with poles arranged as those of a two-pole dynamo) with a solid steel armature of special shape capable of revolving between them. The passing of the current round the field-coils tends to move the armature through a right angle. A lever is fixed to the end of the armature, and the brake, which is normally held on to the brake wheel by a weight, is connected thereto by a rod. The movement of the armature lifts the brake and releases the brake wheel. The great advantage of the particular form of magnet described lies in the strong pull and long range which it affords. The brake is also connected to a lever which stands at the driver's left, so that it can be operated instantly by hand when necessary.

20. The controller is of the so-called Universal type, and con-

sists of two separate controllers combined in one case and actuated by one handle. One controller actuates the lifting, and the other the sluing, and both are connected by pinions and quadrants to the lever. The latter is so arranged, by means of a Universal device, that when moved in a vertical plane it actuates the lifting, and, in a horizontal plane, the sluing controller. Thus it will be seen that the hook follows the motion of the driver's hand, lifting when he raises the lever, and so on. One conspicuous point about the controller is the ease with which it is manipulated, owing to the absence of click gear upon the immediate contacts. Notches are provided only at the two extremes and the off positions, and, in consequence, the controller may be operated throughout the day without fatigue and its attendant slackness. The resistances are fixed inside the cabin.

21. The switchboard consists of polished slate, framed in teak, and fitted with main double-pole switch and automatic cut-out in lifting circuit, and isolating switches in sluing circuit, fuses and lighting switches.

22. The crane cabin is well lighted by fixed and portable lamps, and a cluster, in an enamelled iron reflector, is fixed beneath the jib half way up its length. Terminals are provided upon the switchboard, to which may be connected an inspection voltmeter and ammeter.

23. Each 3-ton crane on being completed was tested in the following manner:—

A large iron tub (fitted with a valve at the bottom), weighing 1 ton net, brought close to the quay wall, lowered into the dock, filled through the valve, there being a port in the side of the tub so that the weight of water could not exceed 2 tons, or a total load of 3 tons. This was lifted 30 feet, and slued through a half circle simultaneously, then lowered on to the quay, where the valve on the bottom of the tank opened, and the water returned into the dock. The tub was again lifted empty, taken back, and the same process continued.

24. Each crane was worked in this manner for three hours, making on an average 100 lifts in the stated time. The current used was:—

Lifting..	18.3 B.O.T. units.
Sluing..	6.4 " "

or a total of 24.7 Board of Trade Units or 2.74 Board of Trade Units per 1,000 foot-tons, or a total cost of 5.4*d* per 1,000 foot-tons, as shown in the Table No. 2 of costs.

25. The 10-ton cranes, Fig. 467, are of the same design as the 3-ton cranes. The revolving motor is 12 brake horse-power at 1,000 revolutions per minute, and the lifting motor, which is of 60 brake horse-power, in single gear drives directly through a pinion on to the barrel shaft, the ratio of gearing being 8 to 1, and a double part of rope is used. The double gear gives a multiplying power of 20 to 1. With the single gear loads up to 2 tons are lifted, and the double gear is used for loads from 2 to 10 tons.

26. As it was impossible to get both the hydraulic and the electric cranes working together under similar conditions into a vessel, on account of the cargoes at Middlesborough being of such a varied nature and the vessels differing so much in dimensions, to obtain comparative figures as to the cost of the work done, a special test was made, which was considered to be the nearest approach to the actual working conditions in loading cargo into a vessel's hold, and the tabulated form shows work done and the power required. The first load was 2 tons lifted through a height of 20 feet by the single gear, slued through 180 degrees, and then lowered 20 feet; the light chain hoisted 20 feet, and the crane slued back through 180 degrees. This cycle of evolutions was performed five times. The load was then increased a further 2 tons each time, up to and including 10 tons, and the same number of evolutions made with each load as in the first case, but double gear being used for the lifting. The total current used was 8.1525 Board of Trade Units, or 2.717 per 1,000 foot-tons, at a cost of 5.4*d.* per 1,000 foot-tons, equivalent to 358,710 foot-pounds of energy per minute for the total work.

27. The author does not propose to take up the members' time by going fully into the details of the hydraulic crane, as this is so well known, but simply contents himself by stating that it was of up-to-date design, as shown in accompanying drawing, Fig. 470, with a capacity of 10 tons, and having three vertical cylinders for lifting. Water is admitted to the center cylinder for the light power, the two outside cylinders for the second power, and all three cylinders for the full load. The sluing is done by gearing driven by a small ordinary four-cylinder capstan-engine, and the working pressure at the crane during the test was 700 lbs. The hydraulic cranes are fifteen years old, but are all of exactly the same design as supplied by the same firm at the present day, and were overhauled and in perfect condition before the trials were made; and the capstan is quite modern.

28. The power-water was measured by means of two Kent's high-pressure water-meters, one on the lifting and the other on the revolving motion, and the crane was worked at its ordinary speed. Precisely the same evolutions were carried out as in the case of the electric crane, and the total gallons of pressure water used was 2,391, equivalent to a cost of 10.25*d.* per 1,000 foot-tons lifted, or equivalent to 687,412 foot-pounds of energy per minute for the total work.

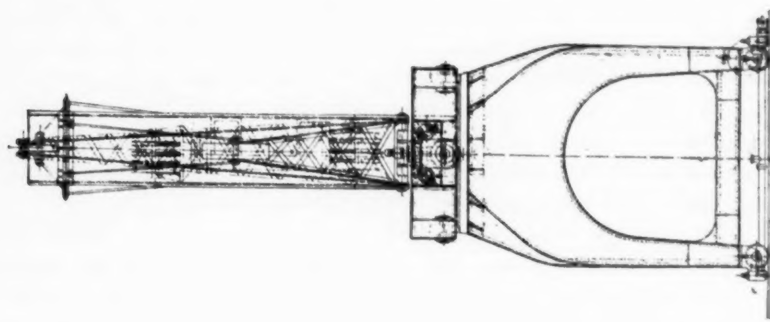
29. Another test was made on a Sunday with six hydraulic cranes arranged to be as near as possible on one quay, and water meters were attached to each crane. Wagons of rails were brought underneath the cranes and worked as if they had been ordinary cargo into a ship. They were lifted to a height of 20 feet, slued half circle, lowered down to empty wagons, the light chain lifted a height of 20 feet, revolved through half circle, and lowered down again. Six hundred tons were dealt with as constant loads, and 600 tons as variable loads, and the tabulated statement shows the cost of handling the same. The cost of power per 1,000 foot-tons of work was 1*s.* 6*d.*, the time occupied seven hours, as shown in Table 1 of hydraulic costs.

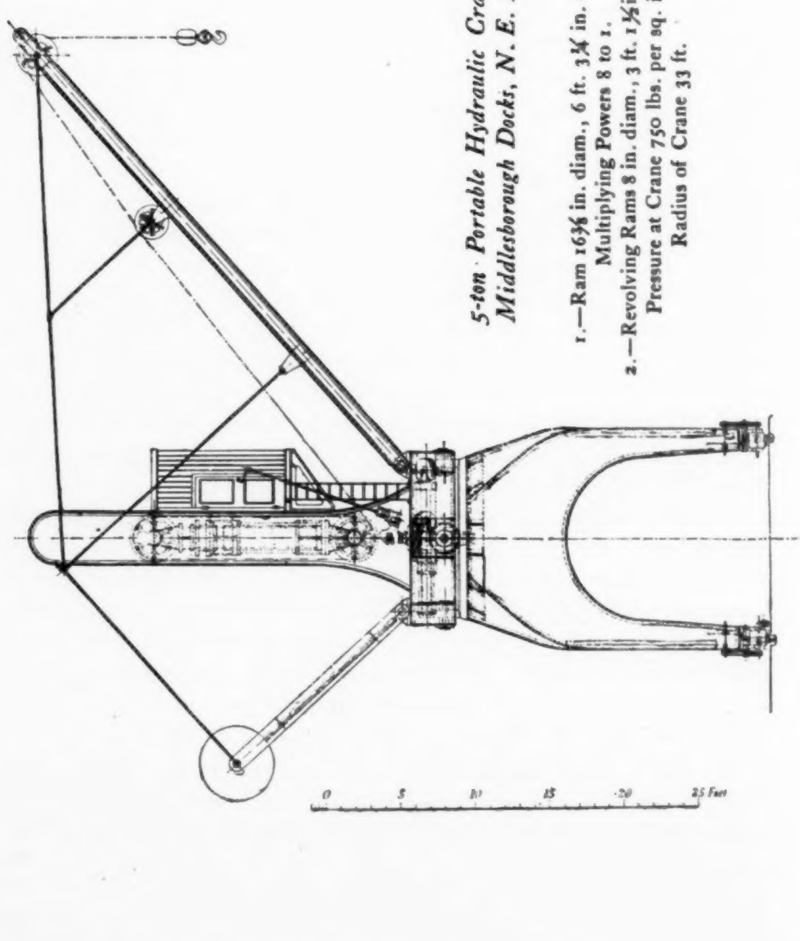
30. The same work was done on the following Sunday by the electric cranes, and the cost per 1,000 foot-tons was 1*s.* 2.23*d.*, and the time occupied 5¼ hours. (See Table 2.)

31. The total saving effected by the electric cranes, including cranemen and laborers, was 25 per cent., but the load factor for the electric was only 7.3 per cent. as against the hydraulic 14.4 per cent., so that, if more cranes had been working and a better load on it at the power station, better results would have been obtained for the electric cranes. The total estimated gain would have been 50 per cent.

32. The diagrams, Figs. 471 and 472, show the fluctuations of the work.

33. *Capstans.*—The electric capstans, Fig. 468, are 24 brake horse-power at 1,000 revolutions per minute, and are capable of exerting a steady pull of 1 ton at a speed of 200 feet per minute, or of hauling a load of 100 tons along a level road. The capstan-head is driven by a cast-iron spur-wheel, which is engaged with a steel pinion. The latter is keyed on the same shaft as a brass worm-wheel. The driving worm is cut from a blank forged solid with its shaft, and is coupled direct to the motor spindle. The worm and wheel work in an oil bath. The motor runs at 1,000





5-ton Portable Hydraulic Crane, Middlesborough Docks, N. E. Ry.

1.—Ram 16⅜ in. diam., 6 ft. 3¼ in. stroke.
Multiplying Powers 8 to 1.
2.—Revolving Rams 8 in. diam., 3 ft. 1½ in. stroke.
Pressure at Crane 750 lbs. per sq. in.
Radius of Crane 33 ft.

FIG. 469.

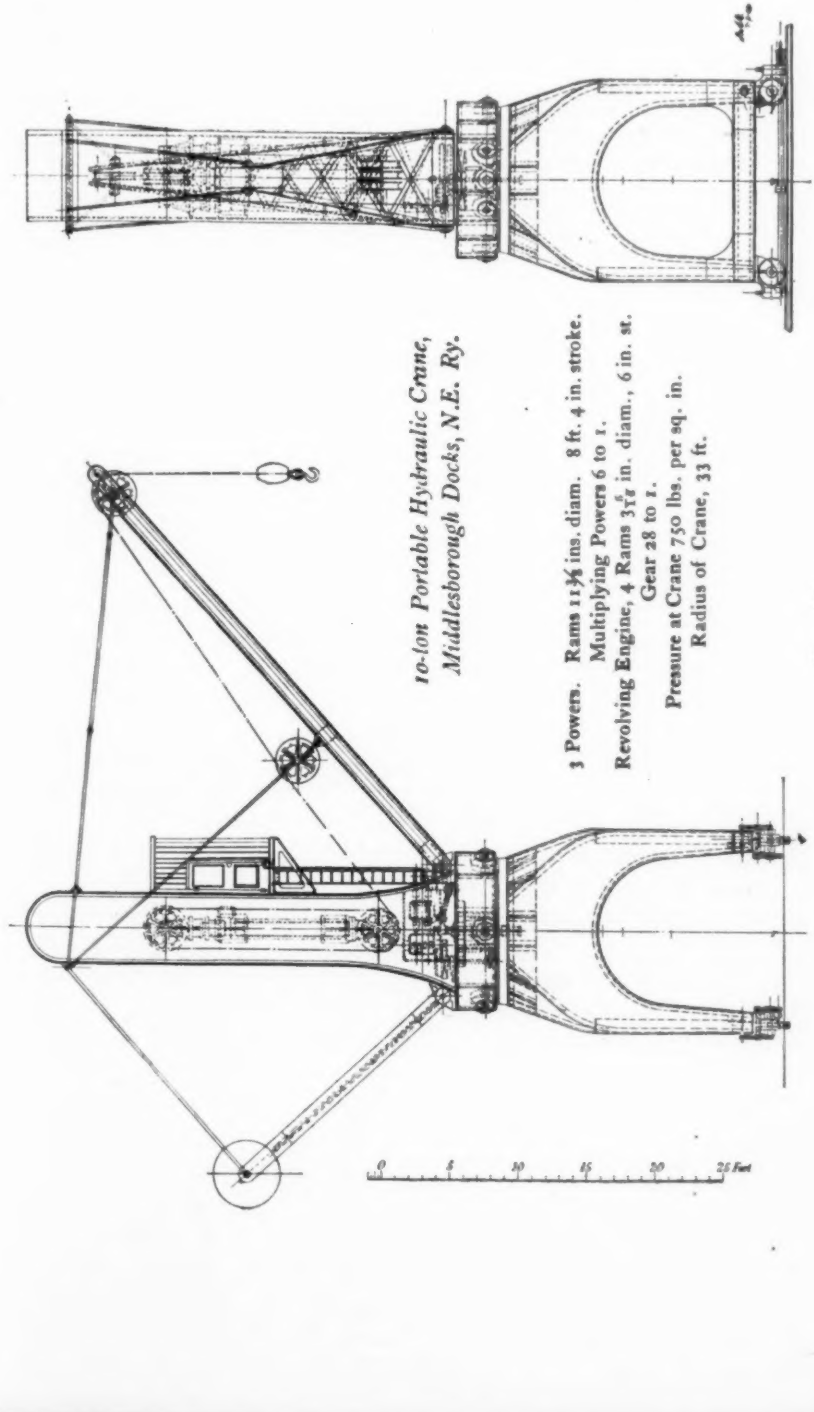

10-ton Portable Hydraulic Crane,
Middlesborough Docks, N.E. Ry.
3 Powers. Rams $11\frac{3}{8}$ ins. diam. 8 ft. 4 in. stroke.
Multiplying Powers 6 to 1.
Revolving Engine, 4 Rams $3\frac{5}{16}$ in. diam., 6 in. st.
Gear 28 to 1.
Pressure at Crane 750 lbs. per sq. in.
Radius of Crane, 33 ft.
0
5
10
15
20
25 Feet

FIG. 470.

Diagram showing Work of Hydraulic Engines, May 24, 1903.

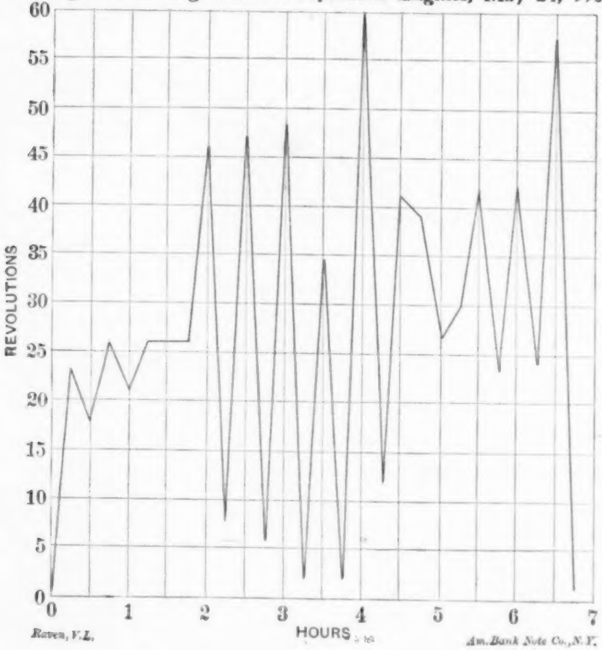


FIG. 471.

Electric Engine Current Diagram, May 31, 1903.

Dotted line shows variations in the current load during the working of Light Loads.
Full line shows variation in the current load during the working of Heavy Loads.

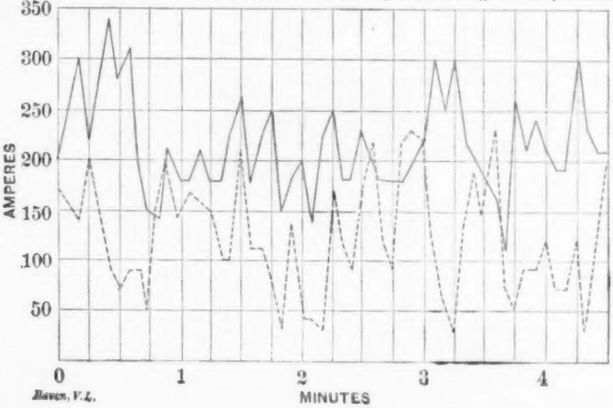


FIG. 472.

revolutions per minute, and is completely enclosed. In construction and design it is similar to the crane motors, but is shunt-wound to avoid large variations in speed of the capstan head. Upon the motor shaft there is fixed an automatic mechanical brake, the principle of which provides that when the capstan-head is driven by the motor the brake releases itself automatically, but should the capstan tend to run back through being overhauled by the weight, the brake at once locks itself and sustains the load. Such a brake is absolutely necessary upon an electric capstan, as the latter differs from a hydraulic capstan in one important respect, namely, that in the hydraulic capstan the water in the cylinders holds the load, whereas in the electric capstan there is nothing to sustain the load upon the current being cut off, and if the load is allowed to run back a serious accident may ensue. It is also essential that the brake should be automatic so as to claim none of the driver's attention from his work.

34. The electrical switch gear presents many points of interest. It consists of a controller with magnetic blow-out, and with overload release gear. The controller is worked by a pedal projecting above the capstan case by about 4 inches. This pedal is removed when the capstan is out of use. The pedal is connected to a dash-pot which prevents the controller being operated too rapidly when a driver is starting the capstan, but, at the same time, by means of valves in the plunger, allows the controller to return rapidly to the off position. The return to the off position is effected by means of a weight which is lifted as the pedal is depressed. In the event of the capstan being pulled up by sudden overload, the release gear works instantaneously and breaks the main circuit. Upon the pressure being removed from the pedal, the return of the controller to the off position automatically replaces the release switch, and the capstan is ready to start again. The whole of the gear is contained in a water-tight cast-iron case sunk in the ground, the top of which consists of checkered plates flush with the quay side. The trials of these capstans are shown in Table 3.

35. *Cables.*—The distribution of current is effected by a network of feeders and distribution cables laid below ground. These cables are fibre-insulated and lead-sheathed; they are laid in wooden troughs on the solid system, and are at a depth of about 2 feet from the surface. Water-tight junction boxes are provided at the junctions of the feeders. The crane connection boxes are

in two portions, the bottom of which contains the positive and negative distributing cables, and the top the connection apparatus for the cranes. The connection socket consists of a circular gun-metal hood which accurately fills the aperture in the cover of the box, and is so shaped as to shield the joint from wet. The interior of this hood contains two gun-metal tongues which meet the two clip contacts in the interior of the box. A stud and groove ensures that the socket shall enter the aperture only in the correct position. The flexible crane cable enters the hood of the socket through a trumpet mouth, and the whole apparatus is shaped so as to obstruct the quay side as little as possible. There are fifty-two boxes in all, placed 20 yards apart. They have withstood the test of the severest weather, and have given no trouble. The cost of cables complete is only about 50 per cent. of the cost of hydraulic mains for the same power.

36. *Lighting.*—About half a mile from the power station a sub-station, fitted with switchboard for the lighting has been put down. There is only one cable for supplying the power both for the electric machinery and the lighting of the docks, offices, and station.

37. *Summary.*—The Traffic Department (see Table 4) find the electric cranes to be a great advantage. There is very little time required for oiling, as the crane has ring lubricators to all main bearings, and the wheels run in oil-baths, which only require attention about once a month. The oil-bath effectually prolongs the life of the wheels and deadens any noise, which is a great advantage to the drivers in hearing instructions. The controlling of the crane by one handle is also a distinct advance both over the steam and the hydraulic cranes. With the steam cranes the driver had to attend to four levers and a footbrake, in addition to the feed-pump and the firing of the boiler. In the case of the hydraulic crane two levers are in use which require considerable force.

38. The automatic cut-out in the electric cranes cuts off the current in one case at 3 tons 5 cwt. and the other at 10 tons 10 cwt., and this device prevents an overload upon the crane. Should the current fail at any time the brake on the lifting gear is applied automatically, and will hold the full load in any position with safety. To obtain some definite idea of the speeds of the various cranes when working under ordinary conditions, they were tested separately. The steam crane had a load of 2 tons put on, and

this was lifted 30 feet, slued through 106.5 feet, lowered 30 feet, light hook lifted 30 feet, slued 106.5 feet, and light hook lowered 30 feet. The total time for these operations was 1 min. 44 secs., or at the rate of 34 cycles per hour.

39. The hydraulic cranes, under precisely the same conditions, occupied 1 min. 40 secs. per cycle, or equivalent to 36 cycles per hour.

40. The electric cranes did the same work in 64 secs. per cycle, or 56 cycles per hour, and this is capable of being increased beyond even this point with a good driver. Therefore, in ordinary working, the electric cranes are doing 50 per cent. more work per hour than the hydraulic or steam cranes. On actual test it was found that the electric cranes can be released from the rails, traveled 30 feet, and refixed to rail in 3 minutes.

41. When it has been necessary to move the hydraulic cranes to suit the working into the various ships' holds, six men have had to be called one hour earlier in the morning to set the four cranes which are required for each vessel. With the electric cranes the men have not to be called earlier, as two men in summer and four in winter can disconnect and connect up four cranes in 15 minutes. In winter, on frosty nights, four men have been employed 6 hours each for the purpose of keeping fires to prevent water in the cylinders of the hydraulic cranes from freezing; on the other hand, the electric cranes have had no need of this. When a steam crane had to be got ready for work, the driver had to commence one hour earlier in order to obtain steam, whereas the hydraulic and electric cranes are ready for work at any moment. In 1903, when the steam cranes were superseded by the electric cranes, the quantity of traffic dealt with at this dock was:

	Tons.
Coal and coke exported	259,746
Merchandise exported	297,304
Merchandise imported	33,696
Total	590,746

42. To do this work, the quantity of coal burned at the new power station, which, as previously stated, supplies both the electric power, lighting, and hydraulic power water, was: 3,428 tons 4 cwt., or the total merchandise worked per ton of coal burned was 172.3 tons, or a gain of 47.2 per cent.

TABLE I.

Work Done by Hydraulic Cranes, Middlesborough Dock, May 24, 1903.

Note.—Each crane lifted each and every load 20 feet, slued through a hook travel of 103.5 feet, lowered load 20 feet, lifted light chain 20 feet, slued back through a hook travel of 103.5 feet, and lowered the light chain 20 feet, thus completing each cycle of evolutions.

No. of Crane.	Capacity of Crane.	No. of Rails per Lift.	No. of Lifts.	Gross Weight of Rails.				Water Meter Reading at Start.	Water Meter Reading at Finish.	Total Gallons used.
				T.	*C.*	*Q.*	*Lbs.*			
1	35 cwts.*	5	70	100	6	1	0	34,280	35,900	1,620
2	5 tons.	7	50	100	11	2	12	12,360	15,280	2,920
3	10 "	7	50	100	16	3	24	660	3,610	2,950
4	35 cwts. *	5	70	100	0	3	16	150	2,170	2,020
5	35 " *	5	70	100	11	2	12	110	2,030	1,920
6	5 tons.	7	50	99	18	0	19	100	3,240	3,140
First Part of Test	Totals.....		360	602	5	1	27	......	......	14,570

* One lifting ram, 10½ inches diameter × 6 feet 3 inch stroke.
Two sluing rams, 5¾ inches diameter × 3 feet and 3 feet 8 inch strokes.
Pressure at Crane 750 pounds.
Multiplying power, 8 to 1.
Height and radius of jib similar to 5 ton hydraulic crane.

Power Station Records and Costs.

No. of Hydraulic Engine at work.	Time Started for Test.	Engine Counter Reading at Start.	Time Finished for Test.	Engine Counter Reading at Finish of Test.
No. 3.	7 A.M.	4,818,441	2 P.M.	4,829,455

Work Done by Hydraulic Cranes, Middlesborough Dock, May 24, 1903.

No. of Crane.	Total No. of Lifts at 2 Rails per Lift.	Gross Weight of Rails.				Water Meter Reading at Start.	Water Meter Reading at Finish.	Total Gallons used.
		T.	*C.*	*Q.*	*Lbs.*			
1	29	17	1	3	4	35,900	36,600	700
2	17	10	0	1	12	15,280	16,290	1,010
3	17	10	0	1	12	3,610	4,630	1,020
4	29	17	1	3	4	2,170	3,010	840
5	29	17	1	3	4	2,030	2,780	750
6	17	10	0	1	12	3,240	4,390	1,150
	138	81	6	1	20	....	....	5,470
		Second Part of Test.				Total No. of Lifts, 465.		

Power Station Records and Costs—(*Continued*).

Total Revolutions made.	Load Factor.	Total Gallons pumped.	Total Gallons used by Cranes as above.	Total Gallons used by Capstan as measured by Water Meter.
11,014	*Per cent.* 14.4	59,002 (displacement)	34,070 (displacement)	14,110

TABLE I.—*Continued.*

WORK DONE BY HYDRAULIC CRANES, MIDDLESBOROUGH DOCK, MAY 24, 1903.

No. of Crane.	Total No. of Lifts at 4 Rails per Lift.	Gross Weight of Rails.				Water Meter Reading at Start.	Water Meter Reading at Finish.	Total Gallons used.
		T.	*C.*	*Q.*	*Lbs.*			
1	29	34	3	2	8	36,600	37,310	710
2	17	20	0	2	24	16,290	17,380	1,090
3	17	20	0	2	24	4,630	5,600	970
4	29	34	3	2	8	3,010	3,820	810
5	29	34	3	2	8	2,780	3,540	760
6	17	20	0	2	24	4,390	5,570	1,180
	138	162	12	3	12	....	....	5,520
Total weight.......		608	2	3	12	Total Gallons used, 19,500.		

POWER STATION RECORDS AND COSTS—(*Continued*).

Total Gallons run through Drains as Measured.	Total Gallons run through Momentum Valve as measured.	Total Gallons accounted for.	Coal burnt during 7 Hours' Test in pounds.	Ashes and Clinker left after Test in pounds.
8,720	2,100	59,000	3,584	330

WORK DONE BY HYDRAULIC CRANES, MIDDLESBOROUGH DOCK, MAY 24, 1903.

No. of Crane.	Total No. of Lifts at 6 Rails per Lift.	Gross Weight of Rails.				Water Meter Reading at Start.	Water Meter Reading at Finish.	Total Gallons used.
		T.	*C.*	*Q.*	*Lbs.*			
1	29	51	5	1	12	37,310	37,990	680
2	17	30	1	0	8	17,380	18,410	1,030
3	17	30	1	0	8	5,600	6,660	1,060
4	29	51	5	1	12	3,820	4,640	820
5	29	51	5	1	12	3,540	4,360	820
6	17	30	1	0	8	5,570	6,770	1,200
	138	243	19	1	4			5,610

POWER STATION RECORDS AND COSTS—(*Continued*).

Gallons pumped per cwt. of Coal burnt.	Gallons pumped per cwt. of Combustibles.	Cost of Coal at 8*s.* 3*d.* per ton.			Cost of Wages in Power House during Test.			Cost of Oil Stores and Water during Test.		
		£	*s.*	*d.*	£	*s.*	*d.*	£	*s.*	*d.*
1,844	2,034	0	13	2	0	8	7.6	0	2	5

TABLE I.—*Continued.*

WORK DONE BY HYDRAULIC CRANES, MIDDLESBOROUGH DOCK, MAY 24, 1903.

Total No. of Lifts at 8 Rails per Lift.	Gross Weight of Rails.				Water Meter Reading at Start.	Water Meter Reading at Finish.	Total Gallons used.	Gross Total Weight of Rails.				Gross Total No. of Gallons used.	No. of Crane.
	T.	*C.*	*Q.*	*Lbs.*				*T.*	*C.*	*Q.*	*Lbs.*		
....	..	..	..	..	......	.....	....	202	16	3	24	3,710	1
17	40	1	1	20	18,410	19,440	1,030	200	15	0	20	7,080	2
17	40	1	1	20	6,660	7,330	670	201	0	2	4	6,670	3
....	..	..	..	..	......	......	....	202	11	2	12	4,490	
....	..	..	..	..	......	......	....	203	2	1	8	4,250	5
17	40	1	1	20	6,770	7,970	1,200	200	1	2	27	7,870	6
51	120	4	1	4			2,900	1,210	8	1	11	34,070	

POWER STATION RECORDS AND COSTS—(*Concluded*).

Total Station Cost during Test.			Total Station Cost per 1,000 Gallons pumped during Test.			Capital Charges during Test (see Statement below).			Capital Charges per 1,000 Gallons pumped during Test.			Total Cost at Station per 1,000 Gallons pumped.		
£	*s.*	*d.*	£	*s.*	*d.*	£	*s.*	*d.*	£	*s.*	*d.*	£	*s.*	*d.*
1	4	2.6	0	0	4.92	1	18	10	0	0	7.9	0	1	0.82

WORK DONE BY HYDRAULIC CRANES, MIDDLESBOROUGH DOCK, MAY 24, 1903.

CAPITAL CHARGES.

Particulars of Capital Charges.

		£	*s.*	*d.*
Value of hydraulic plant, £17,787 19*s.* 10*d.*	@ 5 per cent. interest per annum....... / @ 5 per cent. for repairs and depreciation	1,778	16	0
Value of buildings (proportion), £5,707 0*s.* 0*d.*	@ 5 per cent. interest per annum....... / @ 5 per cent. for repairs and depreciation	570	14	0
Rates and taxes (proportion)..		81	13	4
		£2,431	3	4

			£	*s.*	*d.*
£2,431 3*s.* 4*d.*	÷ 365 days	....................................per day	6	13	2
£6 13*s.* 2*d.*	÷ 24 hours	...........................per hour		5	6.58
£0 5*s.* 6.58*d.*	× 7 hours	...........................time of test	1	18	10
£1 18*s.* 10*d.*	÷ 59,000 gallons	..................per 1,000 gallons			7.9

COST PER 1,000 FOOT-TONS.

Cost of Power per 1,000 Foot-Tons of Work.

Total tons dealt with..	1,210.4 tons.
Total gallons of hydraulic power water used	34,070 gallons.
Total gallons used per 1,000 foot-tons of work	1,407 "
Total cost for power only per 1,000 foot-tons of work...........	£0 1*s.* 6*d.*

TABLE II.

Work Done by Electric Cranes, Middlesborough Dock, May 31, 1903.

First Part of Test.

No. of Crane.	Capacity of Crane.	No. of Lifts at 4 Rails per Lift.	3 Rails per Lift.	2 Rails per Lift.	5 Rails per Lift.	6 Rails per Lift.	7 Rails per Lift.	Total No. of Lifts.	Total Weight of Rails. T.	C.	Q.	Lbs.	Total Board of Trade Units used.*
1	3 Tons.	60	5	5	..	..	..	70	102	9	0	12	75
2	3 "	60	5	5	..	..	..	70	102	9	0	12	
3	3 "	60	5	5	..	..	..	70	102	9	0	12	
4	10 "	..	..	..	40	5	5	50	101	13	3	20	
5	3 "	..	..	..	40	5	5	50	101	13	3	20	
6	3 "	..	..	..	40	5	5	50	101	13	3	20	
Totals		180	15	15	120	15	15	360	612	9	0	12	75

* The current for all the six cranes was measured through a single meter at the end of the jetty cable.

Power Station Records and Costs.

No. of Electric Engine at Work.	Time Started for Test.	Time Finished Test.	Meter Reading at Start of Test in Board of Trade Units.	Meter Reading at Finish of Test in Board of Trade Units.
No. 3.	A.M. 8.30	A.M. 10.45	180,080	180,158
No. 3.	A.M. 11.30	P.M. 2.30	180,158	180,344
Totals......				

Work Done by Electric Cranes, Middlesborough Dock, May 31, 1903.

Second Part of Test.

Total Lifts at 2 Rails per Lift.	Total Weight of Rails. T.	C.	Q.	Lbs.	Total Lifts at 3 Rails per Lift.	Total Weight of Rails. T.	C.	Q.	Lbs.	Total Lifts at 4 Rails per Lift.	Total Weight of Rails. T.	C.	Q.	Lbs.	Total Lifts at 6 Rails per Lift.
30	22	15	1	12	30	34	3	0	4	30	45	10	2	24	..
30	22	15	1	12	30	34	3	0	4	30	45	10	2	24	..
30	22	15	1	12	30	34	3	0	4	30	45	10	2	24	..
17	12	18	..	4	17	19	7	0	6	17	25	16	0	8	4
17	12	18	..	4	17	19	7	0	6	17	25	16	0	8	4
17	12	18	..	4	17	19	7	0	6	17	25	16	0	8	4
141	107	0	0	20	141	160	10	1	2	141	214	0	1	12	12

Summary of Second Part of the Test: Electrical cranes. Total number of lifts, 474. Total weight lifted, 612 tons 9 cwt. 12 lbs. Total Board of Trade units used, 99.

TABLE II.—*Continued.*

POWER STATION RECORDS AND COSTS.

Total Board of Trade Units used.	Total Board of Trade Units used by Lamps in Cellar at Power Station.	Total Board of Trade Units used by Arc Lamps on Lighting Circuit as per Meter.	Net Total Board of Trade Units used by Electric Cranes.	Coal Burnt during Test in Cwts.	Coal Burnt per Board of Trade Unit in Lbs.
78	3	nil	75	12	17.3
186	3	84	99	14	8.4
264	6	84	174	26	....

Load Factor, 7.3 per cent.*

* This figure represents the Load Factor in proportion to the three electric engines working at maximum speed.

WORK DONE BY ELECTRIC CRANES, MIDDLESBOROUGH DOCK, MAY 31, 1903.

Second Part of Test.

Total Weight of Rails.				Total Lifts at 7 Rails per Lift.	Total Weight of Rails.				Total Weight of Rails.				Total Board of Trade Units used.*
T.	*C.*	*Q.*	*Lbs.*		*T.*	*C.*	*Q.*	*Lbs.*	*T.*	*C.*	*Q.*	*Lbs.*	
..	..	..	..	..	..	..	..	..	102	9	0	12	99
..	..	..	..	..	..	..	..	..	102	9	0	12	
..	..	..	..	..	..	..	..	..	102	9	0	12	
9	2	0	16	16	34	10	2	14	101	13	3	20	
9	2	0	16	16	34	10	2	14	101	13	3	20	
9	2	0	16	16	34	10	2	14	101	13	3	20	
27	6	1	20	39	103	11	3	14	612	9	0	12	99

* The current for all the six cranes was measured through a single meter at the end of the jetty cable.

SUMMARY OF SECOND PART OF THE TEST: Electrical cranes. Total number of lifts, 474. Total weight lifted, 612 tons 9 cwt. 12 lbs. Total Board of Trade units used, 99.

POWER STATION RECORDS AND COSTS.

Average Coal burnt per Board of Trade Unit in Lbs.	Ashes and Clinkers left on Conclusion of Test.			Cost of Coal at 8*s.* 3*d.* per Ton during Test.			Cost of Wages Net Time of Test.			Oil Stores and Water during Time of Test.		
	C.	*Q.*	*Lbs.*	£	*s.*	*d.*	£.	*s.*	*d.*	£	*s.*	*d.*
11	2	2	0	0	10	9	0	6	5.7	0	2	1

TABLE II.—*Continued.*

WORK DONE BY ELECTRIC CRANES, MIDDLESBOROUGH DOCK, MAY 31, 1903.

Summary of Tests

Gross Total Weight of Rails.				Gross Total Board of Trade Units used.	Remarks.
T.	*C.*	*Q.*	*Lbs.*		
204	18	0	24	174	Each crane lifted each and every load 20 feet, slued through a hook travel of 103.5 feet, lowered load 20 feet, lifted light chain 20 feet, slued back through a hook travel of 103.5 feet, and lowered the light chain 20 feet, thus completing each cycle of evolutions.
204	18	0	24		
204	18	0	24		
203	7	3	12		
203	7	3	12		
203	7	3	12		
1224	18	0	24	174	

POWER STATION RECORDS AND COSTS.

Total Station Cost during time of Test.			Total Station Cost per Board of Trade Unit.			Capital Charges, Depreciation, Interest, Repairs, Rates, during Test. *See* Summary.			Capital Charges per Board of Trade Unit.			Total Cost per Board of Trade Unit.		
£	*s.*	*d.*	£	*s.*	*d.*	£	*s.*	*d.*	£	*s.*	*d.*	£	*s.*	*d.*
0	19	3.7	0	0	0.87	1	5	9.75	0	0	1.17	0	0	2.02

WORK DONE BY ELECTRIC CRANES, MIDDLESBOROUGH DOCK, MAY 31, 1903.

CAPITAL CHARGES.

Particulars of Capital Charges.

	£	*s.*	*d.*
Rates	81	13	4
Value of buildings, £4,507 { @ 5 per cent. for interest; @ 5 per cent. for repairs and depreciation }	450	14	0
Value of plant, £16,219 { @ 5 per cent. for interest; @ 5 per cent. for repairs and depreciation }	1,621	18	0
	2,154	5	4

		£	*s.*	*d.*
£2,154 5*s.* 4*d.* ÷ 365 days	per day	5	18	0
£5 18*s.* 0*d.* ÷ 24 hours	per hour		4	11
£0 4*s.* 11*d.* × 5.25 hours	time of test	1	5	9.75
£1 5*s.* 9.75*d.* ÷ 264	per Board of Trade unit			1.17

COST PER 1,000 FOOT-TONS.

Cost of Power per 1,000 Foot-Tons of Work.

Total tons dealt with	1,224 9 tons.
Total Board of Trade units used	174 units.
Total units used per 1,000 foot-tons of work	7.1 units.
Total cost of power per 1,000 foot-tons of work	£0 1*s.* 2.3*d.*

TABLE III.

ELECTRIC *versus* HYDRAULIC CAPSTAN TESTS, MIDDLESBOROUGH DOCK, AUGUST 19 AND 20, 1903.

Weight of Load.	Distance hauled.	Time occupied.		Speed per minute.		Power used in units and gals.		Power required theoretically taking tractional force at 23 lbs. per ton.	Actual power used by machines.		Efficiency of machines.		Cost of of Power.		Cost of 1,000 foot-tons of work.	
		Elec.	Hyd.	Elec.	Hyd.	Elec.	Hyd.		Elec.	Hyd.	Elec.	Hyd.	Elec.	Hyd.	Elec.	Hyd.
Tons.	*Feet.*	*Secs.*	*Secs.*	*Feet.*	*Feet.*				*Foot-lbs.*	*Foot-lbs.*	*P. c.*	*P. c.*	*Pence.*	*Pence.*	*Pence.*	*Pence.*
174	100	47½	361	126	17	0.51	45	400,200	1,354,764	807,300	30	50				
145	100	40	115½	150	52	0.51	45	333,500	1,354,764	807,300	25	40				
116	100	37	74	162	81	0.31	40	266,800	823,484	717,600	32	36	4.1467	4.3	6.63	6.88
87	100	35	65½	171	93	0.31	80	200,100	823,484	1,435,200	24	14				
58	100	33½	46	179	130	0.26	65	133,400	690,664	1,166,100	19	11				
29	100	33	29½	182	203	0.21	65	66,700	557,844	1,166,100	12	5				
..	..	226	691½	*mean.* 161	*mean.* 96	2.11	340	1,400,700	5,605,004	6,099,600	*mean.* 25	*mean.* 23	..	..	..	..

Power used by *Electric* Capstan running light for 10 minutes—0.5 unit or— 1,328,200 foot-lbs. Value 0.985 pence.
" " " *Hydraulic* " " " " " " 570 gals. or 10,225,800 " " 7.2 "

NOTE.—These tests were made with ordinary dock wagons in bad weather. The wagons had the usual grease-boxes to axles. Lines very badly laid, but level.

TABLE IV.

SUMMARY OF RESULTS.

Description of Cranes.	Total number of lifts made.	Total weight of rails lifted.	Total time occupied in loading.		Total coal burnt at Power Station during test.	Total cost of working at Power Station during test.	Total cost of working at Station, including Capital, Interest and Repairs, during test.	Total cost of men's wages handling cargo during test.	Total cost of handling cargo.	Total cost per 100 tons.	Total saving effected by electric cranes.	Total saving effected by electric cranes.
		Tons.	*Hrs.*	*Min.*	*Lbs.*	*Pence.*	*Pence.*	*Pence.*	*Pence.*	*Pence.*	*Per 100 tons.*	*Per cent.*
Hydraulic............	825	1210.4	7	0	3,584	167.34	436.74	3,528	3,964.74	327.5	..	..
Electric..............	834	1224.9	5	15	2,912	151 38	351.48	2,646	2,997.48	244.7	..	..
Saving effected by Electric Cranes....	..	..	1	45	672	15.96	85 26	882	967.26	82.8	82.8	25

DISCUSSION.

*Mr. E. B. Ellington.**—Mr. Raven's paper regarded as a record of facts is a most useful contribution to the elucidation of a subject which has roused considerable controversy, and the Institution is greatly indebted to him for it. It is not, however, easy to obtain data which are really comparable, and Mr. Raven's figures and conclusions are open to criticism from this standpoint.

He states in his opening remarks that the hydraulic and electrical appliances at the Middlesborough dock "are working side by side and are all thoroughly up-to-date, giving an excellent opportunity for judging the value of one against the other so far as economy in working is concerned." But this statement can only be applied with accuracy to the generating plant and possibly to the capstan. With regard to the hydraulic cranes it is stated that they were made fifteen years ago. In no sense can these be taken to be comparable with the very perfect electric cranes recently erected.

Fifteen years ago comparatively little attention was paid to the economical production and use of power in docks. The hydraulic system had proved economical as a whole, but there was little competition. The effect has been that a very large number of hydraulic appliances installed are uneconomical as compared with what it is possible to obtain by the system. When electricity for these purposes became available an entirely different set of conditions arose. In order that the electrical system should have a continued success it is essential that the machinery should be constructed in all its details in the most perfect manner, so as to reduce the serious loss of efficiency which would otherwise occur, and also to secure that perfection of control and safety in use which has been the special characteristic of hydraulic plant. Bad examples of hydraulic cranes are extravagant in power and frequently work at slow speeds, but they are safe and not costly to maintain. A bad example of an electric crane is useless. I am not now concerned to show that hydraulic plant is better than electric for dock work. Each has its special advantages, and I have elsewhere expressed my conviction that in the best equipped docks of the future both systems will find a place. Local conditions vary so greatly that a great deal of variation in practice is probably desirable. In some cases

* Member of the Instituti n of Mechanical Engineers.

both powers will be wanted; in others the balance of advantage will be in favor of hydraulic and in others electric, according to the circumstances which may make the special advantages of each system predominate. As to the generating plant at Middlesborough, from the particulars of the trials it appears that the mechanical efficiency of both is practically the same, but the electric plant takes 33 per cent. more coal per unit of energy produced. There is no record of the results of working at lower load factors than 100 per cent., with the exception of those given in the tables which show about the same difference of one-third in favor of hydraulics. The figures are, however, so irregular (17.3 pounds to 8.4 pounds coal burnt per unit) that they are of little value on this point. It is quite certain that at low load factors there is a considerable advantage in economy of coal consumption in the hydraulic system, owing to the nearly constant load on the engine and the variable speed, as compared with the variable load and constant speed of the electric plant. Indeed, Mr. Raven's figures indicate this as the 1,844 gallons pumped per hundredweight of coal form a larger percentage of the maximum of 5,090 than the corresponding figure of 11 pounds of coal per unit is to the equivalent of 21.2 pounds of water per electrical horse-power. But the figure 1,844 is a very poor result, only about 36 per cent. of the maximum. Even under very favorable conditions of load factor and taking long periods of work, it should not be less than 60 per cent. I have obtained about 70 per cent. from one year's end to the other with worse load factors. A few hours test on such a point cannot in my judgment give reliable results, because of the impossibility of obtaining correct data in the time. Mr. Raven's figures are also influenced to a very great extent by the fact that the hydraulic work done was spread over seven hours, and the electric over five and a quarter hours. He, of course, explains how this came about, but the explanation only shows how little up-to-date the hydraulic crane equipment of the dock is. Mr. Raven uses the term load factor in a sense which deprives it of most of its value. The proportion of the actual output to the capacity of the plant installed tells you nothing but the size of the Central Station. As the total electrical power installation is twice the size of the hydraulic, the hydraulic load factor, as Mr. Raven expresses it, is necessarily twice the electric under all conditions of equal total output in a given time. It would be absurd to say that

the results of the trials given on pages 945 and 946 were obtained with a load factor of 33 per cent., yet that is so in the sense in which Mr. Raven uses the term. The load factor as usually understood is the relation between the actual average output during the time to the maximum output at any instant during the time. If the Central Station is well divided into units this may approximate to the ratio of the average output to the maximum output of the units in use. Mr. Raven seem to think that if the experiments had been made with a larger number of cranes and better load factors the saving due to the use of electricity would have been doubled, *i.e.*, from 25 to 50 per cent. Of this there is no proof. The load factor on Mr. Raven's definition depends entirely on the average total output per unit of time. This would increase in direct proportion to the number of cranes at work, and the ratio of the hydraulic and electric load factors would remain constant. I assume, of course, that the number of cranes in use under test of each system are equal. If, however, the station plant actually running, or the maximum demand on the engines actually running is taken as the basis of the load factor, then within certain limits the greater the number of cranes the greater will be the discrepancy between the respective load factors. Thus, suppose the number of cranes in use is doubled. The maximum electric output for the number recorded on the diagram is nearly 350 amperes. As there are no accumulators the maximum for twice the number would be, at least, 550 amperes, the power of one station unit. The total output per hour *must* be doubled; therefore, the load factor is increased by not more than $\frac{350 \text{x} 2}{550}$ or 7 to $5\frac{1}{2}$, or say at most 50 per cent. Now in the case of hydraulics the ratio is quite different. The maximum within the limits of a single pumping unit is the same whatever the total output. The engines will work at full speed for a few minutes in the hour or for many minutes approximately in proportion to the total output. If the number of cranes in use is doubled, the total output is doubled also, and the maximum being the same the load factor is increased in the ratio of two to one or 100 per cent. When the total output becomes so large that several pumping engines or dynamos are required, both systems settle down to a characteristic load factor for each. I may also point out that the load factors of the two diagrams (Figs. 471 and 472) are almost equal, but as one is for $4\frac{1}{2}$ minutes and the other

for 7 hours, and as the electrical diagram perhaps includes the constant lighting load, they are hardly comparable. Am I right in assuming that the arc lamp lighting load (page 966) was *constant* during the three hours? The electrical figures of power costs cannot be correct for the current used for the cranes. For the cranes alone the cost works out at more than three pence per unit. A mere increase in output does not in itself improve the load factor in the general current sense of the term. In London the general hydraulic power supply load factor has remained practically constant at 33 per cent. for the past 15 years, though the supply is now more than 900 (nine hundred) million gallons per annum, and has multiplied 7 times in the interval. The area of supply has increased about three times.

Mr. Raven has failed to realize that the load factor of a hydraulic central station plant engaged in supplying intermittent demands, must always be higher than that of an electric station doing the same work, owing to the influence of the hydraulic accumulator. There is a further point which affects the results on which I hope Mr. Raven will give some additional information. There has been installed at Middlesborough about 1,000 electric horse-power and about 500 hydraulic horse-power. The total boiler plant is thus, say, 1,500 horse-power. On what basis has he made the apportionment of the total cost of the station? So much depends on that and on the apportionment of the total running cost. Does he take into account the cost of the cranes? However, Mr. Raven's relative figures are 2d per unit and 1s per 1,000 gallons, and indicate that the cost of the hydraulic energy delivered into the mains is less than the electric. Under these circumstances one would expect to find that when the hydraulic power is applied to lifting operations this initial advantage would be more than maintained. But, Mr. Raven considers from the results of his experiments that this initial advantage of the hydraulic system not only disappears, but is converted into a disadvantage of at least 25 per cent. The figures given are not sufficiently complete to be able to state positively the efficiences of the machines tested, but from the stated power of the motors in the case of the 3-ton electric cranes, it would seem that the electric cranes have an estimated efficiency of about 60 per cent. at full load, while the trial efficiency at full load (2.74 units per 1,000-foot tons) was only about 28 per cent. The 5-ton hydraulic crane had an estimated efficiency of only 58

per cent., and there are no figures available as to actual results obtained with full load. As the efficiency of hydraulic machines, *i.e.* the ratio of useful work done at full load to power expended, only falls below 70 per cent. in ill-designed cranes of the class dealt with in the paper, it is difficult at first to see exactly the reason for the results which Mr. Raven has experienced. But in table 1, it is recorded that 100 tons was dealt with by one of these cranes (No. 6), with an expenditure of 3,140 gallons of water. A similar crane (No. 2) does the same work with 2,920 gallons. The same work is done by a 35 hundredweight crane (No. 1) with an expenditure of only 1,620, and another 35 hundredweight crane (No. 4) takes 2,020 gallons. Such variable results give a very weak basis to work upon. I have known meters to be wrong, but it looks here as if the cranes were at fault. Moreover, a 5-ton single power crane, lifting loads of two tons cannot show well in such a calculation. It is quite evident that if really efficient cranes suitable for the work had been used, the work recorded, in Table 1, *et seq.*, could have been done with half the consumption. It is curious to note however that on the basis of calculations adopted, the only effect on the power station costs would have been a very unimportant difference in the cost of coal. It would have been interesting to have had the same detail as to the power used in the electric cranes (Table 2), but the total consumption alone is given. It has, I think, been well established by past experience that there is only a small annual saving to be effected by small economies due to using plant of variable power, *unless* it is accompanied by a diminution of maximum demand and diminished station plant.

Probably the most important point to determine in order to form a correct opinion as to the relative economy of hydraulic and electric dock plant, is the relative power required at the central station. When once that has been installed, the cost per 1,000 gallons, or the cost per unit will rise and fall with the total output over a given time, but the total cost will remain approximately equal whatever the output, within the limits of the minimum and maximum demand in the same time. The economic advantage which Mr. Raven's figures show for the electric system would disappear if hydraulic cranes had been tested which were of the same maximum power as the electric, and if they had been constructed to do the work in the same time with reasonable efficiency. Speed of working is only a question of requirements,

though probably hydraulic cranes and lifts can be worked at a higher speed than electric without sacrificing safety. In the case of the Middlesborough docks the best possible use appears to have been made of the electric energy transmitted, while the same cannot be said of the hydraulic energy, and in my judgment no conclusions of a general character can with any advantage be drawn from them as to the relative economy in the use of the two systems.

*Mr. John Barr.**—Referring to page 951 of the paper, I might ask whether when lifting the empty hook with the heavy power of 10-ton hydraulic crane, the whole three rams were used? If so, then the efficiency would naturally be small in such case.

Referring to the same crane at foot of same page, the sluing is done by gearing driven by a small ordinary 4 cylinder capstan engine. Had the sluing been done by sluing cylinders, the result would probably have been much more economical.

Might I ask whether in making up the Capital Charges, page 255, the sums taken include the whole plant in including value of the cranes and capstans themselves, or if the sums include only the cost of generating plant in both cases.

It would add to the value of the paper if the approximate first costs were given for hydraulic and electric cranes of the same power; also of capstans.

There is the vital question of maintenance of cranes and capstans. Presumably the electric cranes have been in use for a very few years, while the hydraulic have been fifteen years in use, and evidently in first-class condition even now. It may be too soon to give data, but it would be interesting to know relative costs of maintenance.

If the hydraulic capstans are of the twin-turn-over type, they are very easily got at for repairs. The electric capstans do not appear to be of the turn-over type, and may be rather awkward in case of repairs. There is no reason, however why they should not be so made.

Not having had an opportunity of seeing the electric cranes described at work, the matter of general smoothness and freedom from vibration and noise cannot well be gauged from description given in the paper; but it would be interesting to know how the action of the two types of cranes compare generally, and, if the

---

* **Member of the Institution of Mechanical Engineers.**

electric cranes give the same smooth, quiet, unobtrusive work as hydraulic is so well known to do.

At the middle of page 961 we are told it takes six men to move the hydraulic cranes into position for work in the morning. Why ought this to be? In other words, why furnish the electric cranes with power motors for travelling purposes at lavish cost, and deny the same privilege to hydraulic? Hydraulic cranes can be and are moved along quite readily by a hydraulic engine and gearing.

So far as the tests under varying loads are concerned, the comparison would seem to be a very fair one. Indeed an analysis of the tests shows, I think, that the average of the tests gives 50 per cent. of full load for electric cranes, and about 58 per cent. for hydraulic cranes, which is, of course, in favor of electric in the comparison. It is well known that the principal advantage of electric power is found with light loads, the power given out being approximately in proportion to the load being lifted. On the other hand, the electric plant at generating station must necessarily have large margin to meet sudden demands, and in consequence a smaller load factor, this is a *matter of necessity* and *not* a set-off to the advantage of electric power as hinted at in the paper. On the other hand, the hydraulic pumping plant when at work is working against full load, pumping into the accumulator, even although the pumps may be working only a small proportion of their time. It follows in consequence that an electric power station must have a much larger reserve of power than in the case of hydraulic, as the hydraulic accumulator acts as a reservoir into which the engine can be pumping, while, for the time being, no water may be in course of being drawn off for cranes.

The author of the paper having made the study of electric power as applied to dock appliances his own, he may be pardoned for being perhaps a little prejudiced in favor of electric power. This preference appears in various parts of the paper, notably at foot of page 955, where, after demonstrating—to his own satisfaction at least—that the total saving effected by the electric cranes was 25 per cent., goes on to " estimate," in his own hopeful way, " that the total gain would have been 50 per cent." if certain conditions had been complied with. But the fact is that the question of maintenance—neglected in the paper, might make all the difference. Then again, why should the author in Table 3 go out of his way to point out the immense advantage of

electric power when running a capstan light. He might with equal justice have put it the other way about and quoted the opposite instance of the greater efficiency of hydraulic power with the capstan heavily loaded.

That there is a great field for electric power in dock work, especially in countries where frost prevails, no one will dispute. Both kinds of power, hydraulic and electric, have their own special field. Many, however, will still be unconvinced—the figures in the paper notwithstanding—that taken on all fours there is any gain in economy to be found by using electric in preference to hydraulic power.

The thorough, painstaking and careful manner in which the facts for both electric and hydraulic power for dock work have been put before the meeting in this excellent paper, which must prove to a great extent a standard for future reference, warrants our heartiest appreciation and best thanks.

Mr. Alfred Saxon * said it seemed to him that the critics had taken an English view of the question, and not an American view. It was not a question merely of the saving between electric power and hydraulic power. It was the saving by the dock authorities and by the railway authorities and by the ship owners. Too much had been made of the newness of the electric plant in comparison to the hydraulic plant, which was old and not up to date, but other important features of saving had accrued.

Mr. William Ferguson,* of Wellington, New Zealand, wrote that he had read the paper with much interest. The following points seemed to him to show that the hydraulic tests were not quite what they ought to have been. Out of 59,000 gallons pumped only 34,070 were used for cranes, and 14,100 gallons for capstans, whilst $22\frac{1}{2}$ per cent. of the power generated was wasted; 8,700 gallons going to waste through drains, and 2,100 through momentum valve. This would seem to point out either that the engine power was excessive for the load and that to keep the engines moving it was necessary to run to waste through engine relief valves and accumulator momentum valves, or that the engines were not in an efficient state of repair. In either case the result would probably mean an enhanced cost per 1,000 gallons of hydraulic water utilized. He noticed that cost of repairs and maintenance was set down at 5 per cent. for both electric and hydraulic installations. If the cost of electric plant repairs and

* **Member of the Institution of Mechanical Engineers.**

maintenance were raised by $1\frac{3}{4}$ per cent. to say $6\frac{3}{4}$ per cent., the capital charges would equal that set down for the hydraulic plant, notwithstanding that a larger item for building was charged against hydraulic (probably correctly so as the accumulator house would be additional to the ordinary engine and boiler house buildings). No doubt electric plant would cost more than $6\frac{3}{4}$ per cent. to maintain as compared with 5 per cent. for hydraulic. Though drawings were given of the 5 and 10-ton hydraulic cranes no particulars were given of the 35-hundredweight cranes. The implication was that they were similar to the 5 and 10-ton cranes, that is of the Tannett Walker type, with large roller path. He had always understood that this type gave difficulty in connection with the slewing gear. Whether this was so or not, in this case the three 35-hundredweight cranes did not sem to have been in equally good condition of service. The following abstract of the volume of water used doing equal work is taken from the paper:

	Crane No. 1.	No. 4.	No. 5.
Gallons, page 962	1,620	2,020	1,920
" " 962	700	840	750
" " 933	710	810	760
" " 963	680	820	820
	3,710	4,490	4,250

That similar cranes doing identical work vary in consumption from 3,710 to 4,490 gallons or over 20 per cent. was inexplicable, and this matter should be cleared up. Whilst in the electric tests five sets of experiments were tried with 3-ton cranes and one with a 10-ton crane, in the hydraulic tests two out of the six series were tried with a 5-ton crane, whose trial load in the first test was only 40 per cent. and in the second trials averaged only 29 per cent. of the nominal power of the crane. Taking the 35-hundredweight crane tests alone, which had a reasonable factor of working to total capacity, the volume of water used would be reduced from 1,407 gallons to 1,022 gallons per 1,000-foot tons, which would bring the cost down from 1s. 6d. to 1s. 1d. per 1,000-foot tons, and this notwithstanding the great discrepancy in the volume of water used by the three cranes under trial and consequent doubt thrown on the results.

*Mr. Raven.**—Mr. Ellington seems to raise objection to comparing hydraulic cranes which were put down 15 years ago with

* Author's closure under the Rules.

electric cranes of modern construction; the whole of these cranes, except one, are of exactly similar design to those being erected at the present date; the remaining one, is of 10 tons capacity, and is certainly an up-to-date crane so far as lifting power is concerned; it has three powers, fitted with three cylinders. The revolving motion is obtained by a rotary engine for the reason that the traffic department require the crane to revolve right around when necessary without returning over the same ground. All these cranes were in perfectly good repair.

I quite agree with Mr. Ellington that there are uses to which each class of machinery is specially adapted in dock work; all overhead cranes, elevators, etc., for constant work should, in my opinion, be electric, but for underground machinery, such as dockgate capstans, etc., where there is damp to contend with, hydraulic machinery may possibly have an advantage. In having electric machinery, the same power station can supply light for the docks, stations, offices, etc., and this has many advantages.

Mr. Ellington takes exception to the load factor of the engines when working, but it is simply shown as the work done by each engine in proportion to the maximum capacity of each installation. He also criticises the remarks made, that if a better load factor had been obtained for the electric engines the saving due to electricity would have been doubled.

From observations made at the power station, I find that very little more coal is required to run the engines at a heavy load than that required for a light load. The first and second parts of the electric crane tests power station costs and record sheets will prove this if referred to, as they show that, whereas it took 12 hundredweights of coal to generate 78 units at light loads, only 2 hundredweight more were required to produce 186 units or 156 per cent. more power at the heavy loads, and this would go on reducing the coal consumption until the maximum of the engine was reached.

The arc lamp lighting was constant during the three hours' tests mentioned.

The power station and buildings in connection therewith have been apportioned in proportion to the area required for each installation. Each installation was a separate contract; two boilers were included in the hydraulic contract, and four in the electric. The cost of the cranes was not taken into consideration at all.

With regard to the efficiency of two machines of equal lifting

power, I made a separate test of the three power 10-ton crane as against the 10-ton electric. Each crane made five complete cycles with a 2-ton load, then a further series of five evolutions with 4, 6, 8 and 10 tons, and in dong this the hydraulic crane required 38,495,100-foot pounds of energy, whereas the electrical crane only required 21,656,300-foot pounds, all conditions being equal.

We find that with the electrical cranes the quantity of work dealt with per ton of coal burnt has increased from 117 to 172 tons, or a net gain of 47 per cent., and this is the point we have to consider.

With regard to Mr. John Barr's queries, only the center ram was used for lifting the light hook.

With the 10-ton crane, as already explained, there is a special reason for employing an engine for the revolving motion. If cylinders were used, there would no doubt be a saving of 20 per cent. in water, but you would not be able to revolve right around, and this our traffic department considers necessary.

The capital charges on page 255 only refer to the cost of the generating plant in both cases.

So far as we have at present been able to discover there is practically no difference in the cost of maintenance as between the electric and hydraulic, but I think it is too soon yet to give data. There is very little difference in the general smoothness of working; the gearing of the electrical cranes being machine cut and running in an oil bath.

Neither the hydraulic nor electrical capstans are of the turn-over type; we have a strong objection to them, as we have had several instances where they have come open and serious accidents have happened, in some cases limbs having been broken.

With regard to the moving of hydraulic cranes, even with engine and gearing they cannot be travelled so quickly as electric on account of the heavy walking pipes.

We find in practice that as we have no accumulator in the electrical system that 50 per cent. more engine power is required at hand to meet sudden demands than is necessary for the hydraulic plant, but, on the other hand, the cost of hydraulic engines, accumulator, and additional buildings must be set off against this, and probably more than counterbalances it.

Mr. Barr fails to recognize that capstans are run light for at least 50 per cent. of their working time, therefore the point certainly required showing.

I am sorry that I should have conveyed to Mr. Barr's mind that I was prejudiced in favor of electric power.

I have had a great number of years experience on the North-eastern Railway with all kinds of hydraulic dock machinery, and considered that we had at Middlesborough an exceptional opportunity of comparing electrical with hydraulic machinery for dock work. I therefore instituted a number of tests as near as possible under similar conditions, which have been placed before the institution, and I have judged only upon the results of these trials, which I hope may be found of value to some other as well as my-self.

Mr. Ferguson points out that 22½ per cent. of the power generated was wasted; on the day the tests were made no other work was being done, and consequently the engine power was excessive for the load, and to keep them moving and obtain the best results the water in excess of crane requirements was run to waste; but he seems to have entirely lost sight of the fact that the most economical point of working is when the engines are running at full load; and that, as the load is reduced, the cost of power water increases pro rata; and therefore, so far from the running of the engines being detrimental to the cost, it was an advantage, as the cost per 1,000 gallons was reduced thereby. The cranes were only debited with the cost of water actually used. The engines were practically new and in excellent order.

With regard to the remarks in connection with the percentage of loads to the maximum power of cranes, on looking into this matter, I find that though Mr. Ferguson points out the average load lifted by the 5-ton hydraulic cranes was only about 34 per cent. taking both tests together, he fails to point out that the load factor of the 35-hundredweight hydraulic crane for the total tests average 73 per cent., and against this the electric cranes of 3-tons capacity only average 47½ per cent., and this works out in favor of the hydraulic cranes.

No. 1044.*

REFUSE DESTRUCTION BY BURNING, AND THE UTILIZATION OF HEAT GENERATED.

BY C. NEWTON RUSSELL, BOROUGH ELECTRICAL ENGINEER, METROPOLOLITAN BOROUGH OF SHOREDITCH, LONDON.

(Member Institution of Mechanical Engineers.)

1. The important question of how best to destroy towns' refuse or garbage is now occupying the minds of many engineers and public officials; and all who are responsible for the public health and sanitation are agreed that where possible the refuse should be destroyed by fire. This opinion is evidenced strongly by the fact that the old systems are disappearing, and already most towns of importance have plants for burning garbage.

2. The problem of disposing of towns' refuse has exercised the minds of those connected with the municipal affairs for many years. The old system of tipping it on vacant plots of lands is doomed, as it is now generally recognized that in order to deal with the problem in a really satisfactory manner, other points must be considered besides ease and cheapness of disposal. The most important requirement is that the refuse should be disposed of in a sanitary manner; and whatever method of destruction is decided upon, this should take first place in its consideration. In this paper the author deals only with domestic, or house refuse, which consists of house and shop sweepings, kitchen refuse, market sweepings, and offal, and with what is known as trade refuse; he does not refer to the disposal of sweepings from macadamized or wood-paved roads, except when the sweepings are in a dry condition.

3. The kinds of refuse just enumerated, if left for a few days, ferment and quickly become not only offensive, but dangerous to health. Those who have more than ordinary opportunities for

* Presented at the Chicago meeting (May and June, 1904) of the American Society of Mechanical Engineers, and forming part of Volume XXV. of the *Transactions*.

observing the nature of the materials that go to the dust heap will agree that the only satisfactory way to deal with town refuse is to burn it as quickly as possible. It is much more expensive to burn refuse than to shoot it on the land; but the hygienic benefit derived by the latter method is of such evident and incalculable value to the public health that the matter of cost is of secondary importance.

4. The subject of destructors arranged to dispose of refuse appears to have first been dealt with seriously by Mr. Alfred Fryer more than twenty years ago; in fact it may safely be said that the destructors of to-day are the result of that gentleman's foresight. A very interesting paper on "Dust and Ashes" was written by him as far back as 1887, and shows that he was then fully alive to the possibilities and future usefulness of the destructor. It describes the trouble encountered in dealing with refuse even twenty-five years ago, and the difficulties met with then are the same as we now have to deal with in a more acute form. Mr. Fryer, in order to prove the old saying that there is nothing new under the sun, reminds his readers that in remote times the fires of Tophet received the burnable refuse of Jerusalem.

5. The general arrangement and efficiency of steam-raising destructor plants, as we know them to-day, have considerably improved during the past eight years. In 1897 plants were in operation at Oldham, Warrington, Liverpool, Leyton, Cambridge, etc., and in several parts of London; since then the number has increased considerably. The results from these installations show conclusively that:—(1) In England the refuse has sufficient calorific value to enable it to be easily burned without the addition of coal or any other kind of purchased fuel; (2) The gases produced contain a great amount of energy in the form of heat which can be utilized; (3) The refuse of each locality has a fairly constant calorific value.

The advantages or disadvantages of the various forms of destructor furnaces that have been erected have been pretty well thrashed out by other writers. Each maker claims some advantage over his rivals, but the fact remains that all successful destructors have furnaces operating on practically the same principle, the only difference being in details. Similar results are obtained from the destructors erected by different makers, each claiming superiority.

6. The main object of this paper is to treat the practical side of the question rather than the various types of destructors, and as

full particulars and statistics may be derived from printed publications, the author proposes to describe fully the works with which he is associated. The plant in question is owned by the Metropolitan Borough of Shoreditch, London, and was opened in June, 1897. The works were about the largest of their kind, and contained many new features, some of which even now are subject to criticism. Before the works were built most of the domestic and market refuse was carried in barges by the canal which passes through the Shoreditch area and dumped on waste land in close proximity to the London suburbs. The problem of dealing with the refuse under the new conditions was entrusted to Mr. E. Manville, of the firm of Messrs. Kincaid, Waller and Manville, consulting engineers of Westminster, and the contractors, who erected the destructor, were Messrs. Manlove, Alliot & Co., of Nottingham.

7. The Shoreditch area which adjoins the City of London contains 640 acres, and had in 1897 a resident population of 124,000. Land was very valuable, and it was necessary to purchase and pull down all property near the center of the area in order to obtain a suitable site. Eventually a suitable place was purchased, upon which was erected an electric generating station, refuse destructor, baths and wash-houses, and a free library. The area allotted to the destructor works was 13,450 square feet, costing about 13s. per square foot. The space at disposal being so limited, it was impossible to have an inclined roadway, up which the wagons loaded with refuse could be taken to be tipped directly on the top of the furnaces, which is the usual practice. It was therefore decided to put in lifts, to raise the refuse to the top platform. The destructor and boiler-house, Fig. 473, is separated from the electric generating station by fire-proof doors.

8. The destructor house contains two batteries of three Babcock Wilcox water-tube boilers, each having two refuse furnaces (one placed on either side of the boiler), while the ordinary coal fire-grate immediately under the boiler tubes is provided as an auxiliary for use if required. A complete set thus consists of one boiler and two furnaces, Fig. 474. The hot gases from the refuse furnaces pass through short side flues and immediately come in contact with the boiler tubes. The gases, after passing round the boiler tubes, find their way into one of the main flues, then to the economizer, and thence to the shaft. The grates of the refuse furnaces are 5 feet wide and 5 feet from front to back, and are

inclined at 25 degrees to the horizontal. The refuse furnace bars are of the stationary type, and are built up of wrought-iron wedge-shaped strips in sections 2 feet 6 inches long by 3½ inches wide, and about 4 inches deep. The grate-area of each boiler furnace is 27 square feet, the heating surface being 1,300 square feet.

9. The refuse, when received at the works, is shot into one of two lifts, Fig. 477, each provided with a tipping-truck, Figs. 475

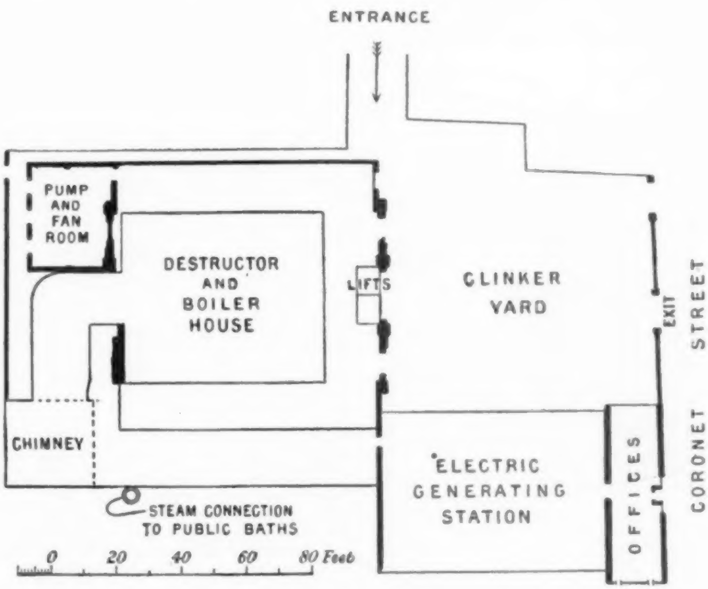


Fig. 473.—Ground Plan of the Works.

to 477, which is run off the lift as soon as it reaches the top platform, Fig. 478; the truck runs upon rails and is operated on the trolley system. The refuse is then tipped into special charging trucks, Figs. 475, 476 an 478, worked by chain gearing, one of which is provided for each furnace.

10. The average amount of refuse received per day is about 85 tons (one ton = 2,240 pounds), and this is delivered between 9 a. m. and 5 p. m., but the amount varies considerably; in summer it may be as low as 60 tons, and in winter as high as 140 tons. As it is found impossible to get the supply at a regular rate, it is advisable to provide storage space to cope with extra large deliveries. This is done at Shoreditch, where a large rectangular

iron storage bin, Figs. 475 and 476, holding about 60 tons of refuse, is fixed under the tipping platforms.

11. The lifts and tip-trucks are worked entirely by electricity, and are found to act well. The average amount of electric energy consumed by the lifts and trucks, taken upon a total of 25,000 tons of refuse (over a year's working), is 0.52 kilowatt-hour per ton, the greater portion of which is expended on the lifting operation. The record time taken to deal with a load of refuse shot

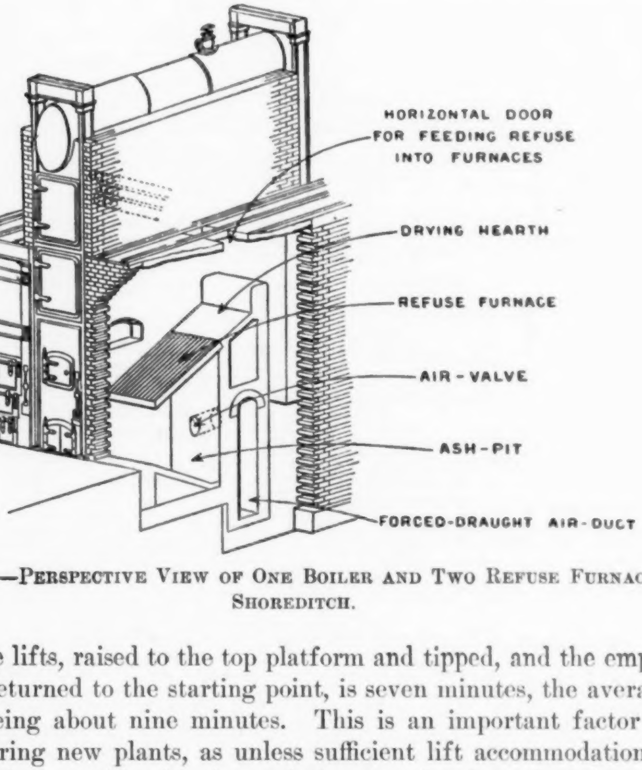


FIG. 474.—PERSPECTIVE VIEW OF ONE BOILER AND TWO REFUSE FURNACES, SHOREDITCH.

into the lifts, raised to the top platform and tipped, and the empty truck returned to the starting point, is seven minutes, the average time being about nine minutes. This is an important factor in considering new plants, as unless sufficient lift accommodation is provided a considerable loss of time results from keeping the dust vans waiting. It is found that the vans in which the dust is collected usually arrive in batches, and unless ample provision is made for rapidly shooting the garbage delay occurs. The cost of repairs to the lift and tipping-truck part of the equipment is very small.

12. The charging trucks are of the square-box pattern, made of mild steel, of sufficient capacity to hold one cartload of refuse,

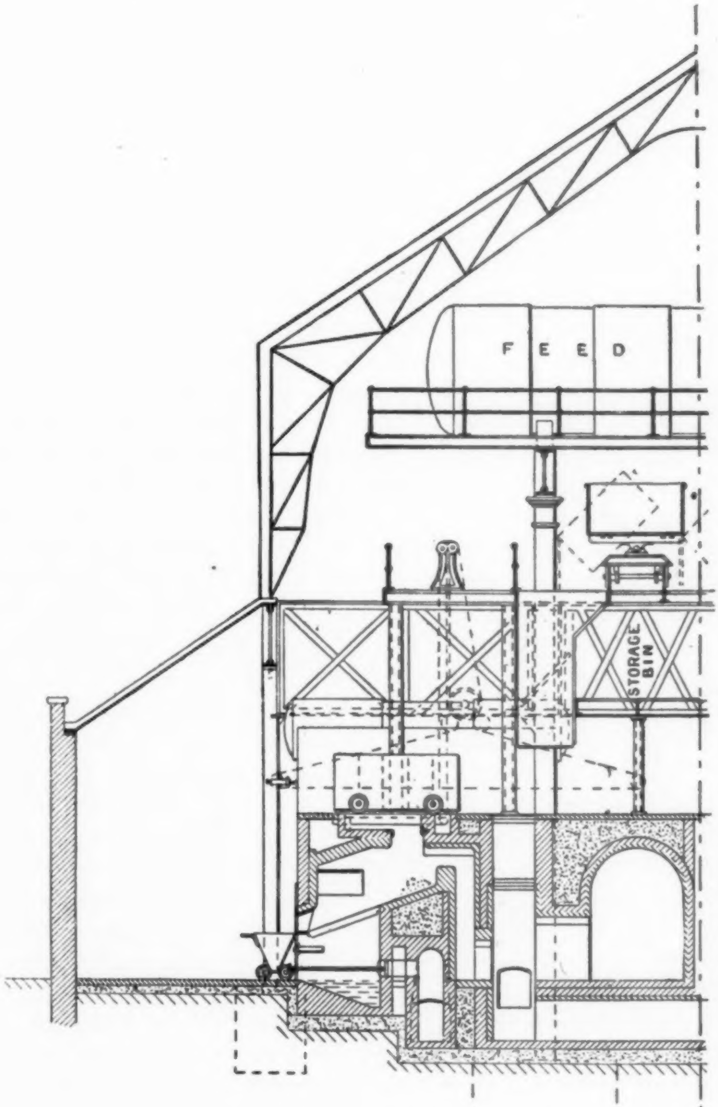


Fig. 475.—Transverse ½ Section.

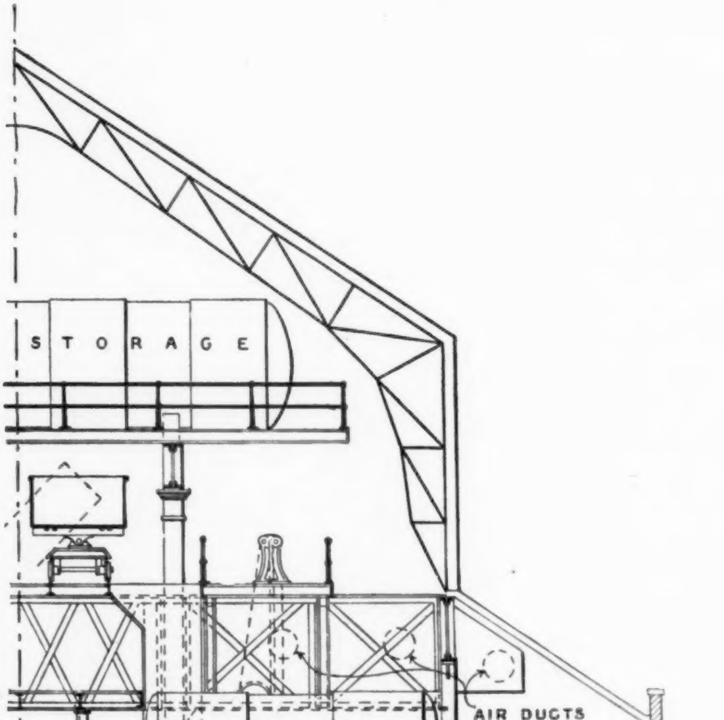


FIG. 476.—TRANSVERSE ½ SECTION.

and are divided vertically into five compartments. Each compartment holds about 500 pounds, the usual amount of one charge of the furnace. The charging-door, or opening, Fig. 478, is of the horizontal pattern, and is operated from the top platform by chain-gearing, worked by hand.

13. To charge a furnace the truck is drawn into position so that one of the compartments is immediately over the horizontal door; the act of opening allows the false bottom of the truck to fall, and the refuse drops into the back portion of the furnace, Fig. 475. However, as the refuse is not sorted in any way, but is dealt with exactly as it is received in the van, it occasionally happens that large articles, such as bags, boxes, tins, etc., prevent the whole of the charge from falling at once on to the drying-hearth, Fig. 474, in which case it has to be pushed down with rods. The charging-trucks enable ordinary refuse to be passed from the carts to the furnace without handling, and are of the kind in use at Liverpool and other places which are worked on Messrs. Boulnois and Brodie's system. It is not found advisable to employ this system of charging for straw, paper, sawdust, or any such light and inflammable material, owing to its tendency to catch fire during the operation of charging and before it can be shot into the furnaces. It is found more convenient and safer first of all to shut off the air-blast and feed the refuse in question through the front of furnace, the blast being turned on again when the grate is well covered. Each furnace is provided with both steam-jet and forced-air blast. In the author's experience the latter has been found preferable.

14. The forced-air blast is provided by three Sturtevant fans, each designed to give 8,000 cubic feet of air per minute, and driven direct by a shunt-wound electric motor at a speed of 650 revolutions per minute. The three fans are not used continuously, the full number being only put to work during the time of heaviest load. The inlets to the fans are connected with air-ducts, Fig. 476, that draw the hot air from the top platform of the destructor; the three fans are connected to a common discharge-duct, which is led underground to the ash-pits, the draught to each being controlled by a separate air-valve, Fig. 476. The air-pressure in the discharge-duct at the fans is 3 inches head of water, while that in the ash-pits is slightly less than 1 inch. Even with the latter comparatively low pressure the temperature obtained in the furnaces often exceeds 2,000 degrees Fahr. It is interesting to

note the comparatively great amount of energy (4 units per ton) absorbed by the electrically driven fans, a matter which is open to improvement. The works are run on the 8-hour-shift system, seven days per week, so that some of the fans are always at work; in fact, they run for weeks without a stop. A direct-coupled, electrically driven fan is an ideal arrangement from a mechanical point of view, and one which experience shows to require very little attention.

15. In order to give an indication of the temperatures reached in steam-raising destructor plants, the author has collected from time to time a number of articles which have passed through the furnaces, from which it appears that such metals as cast-steel, cast and wrought iron, copper, brass, etc., frequently reach a molten state. The furnaces have now been in daily use for six years, and it is interesting to find that the fire-bars are even now in good condition, and show very little sign of deterioration.

16. The fire-brick linings of the furnaces are found to last well, and do not often require renewing; repairs are occasionally necessary in connection with the fire-brick arches, principally on account of damage done by stoking irons. A somewhat novel system of storing hot feed-water is in use (sometimes called thermal storage). The reason that led to its adoption was that, at the inception of the combined scheme, it was not thought that for some time at least the demand for electric current would be very great, and that during the hours of daylight, when refuse was of necessity being burned, it was feared that some of the heat would be lost, owing to the small demand for steam.

17. It was therefore decided to instal Halpin's Thermal Storage system, with a view of storing up heat in the form of water at a high temperature, Figs. 475 to 477. The storage vessel was originally connected to the main line of steam-pipes, with a view to it being gradually filled by pumping in cold feed-water, which would be heated by means of spare heat generated in the refuse furnaces during the daytime. The intention was that a portion of the steam raised in the boilers of the destructor plant would be used as required for the engine driving the electric generators, and the balance passed over to the thermal storage vessel, where it would part with its heat to the feed-water. The engines were designed to have adjustable cut-offs, so as to drive their full load with pressure varying from 200 pounds to 120 pounds per square inch. The object aimed at was to fill up the thermal storage

vessel at times of low load with water at a temperature corresponding to a pressure of 200 pounds per square inch, and at the time of greatest demand for current, not only feed the boilers from

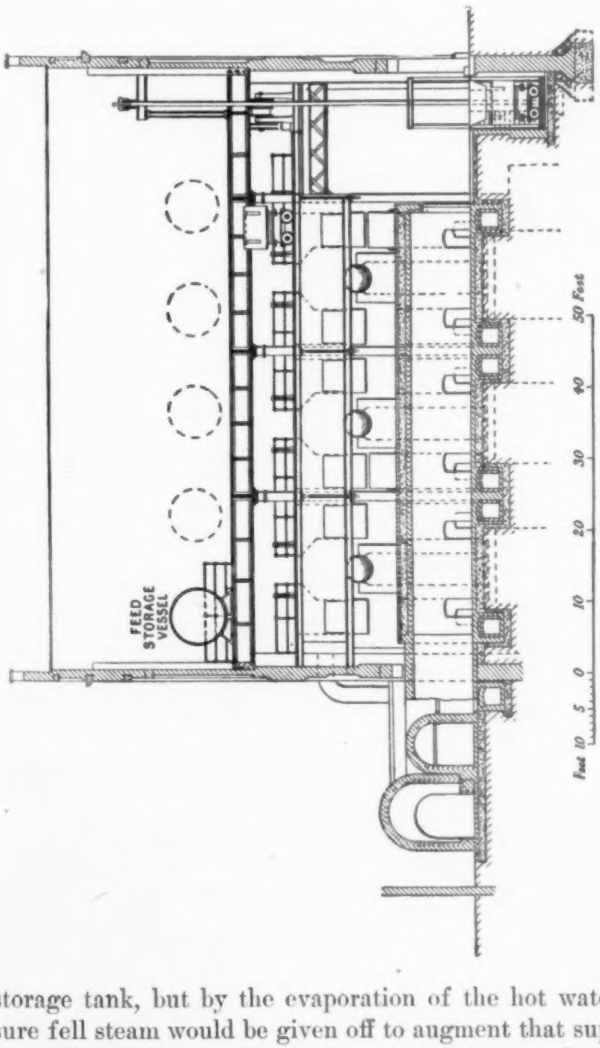


FIG. 477.—LONGITUDINAL SECTION.

the storage tank, but by the evaporation of the hot water as the pressure fell steam would be given off to augment that supplied by the ordinary boilers. The process would proceed until the pressure reached the lowest point, *i.e.*, 120 pounds, by which time it was thought that the time of maximum demand would have passed, and that the ordinary boilers would be able to cope with

the load. The dotted circles shown on Fig. 477 represent positions provided for additional vessels if required. This accounts for the provision of heavy structural ironwork, which has since proved unnecessary, also the extensive system of steam

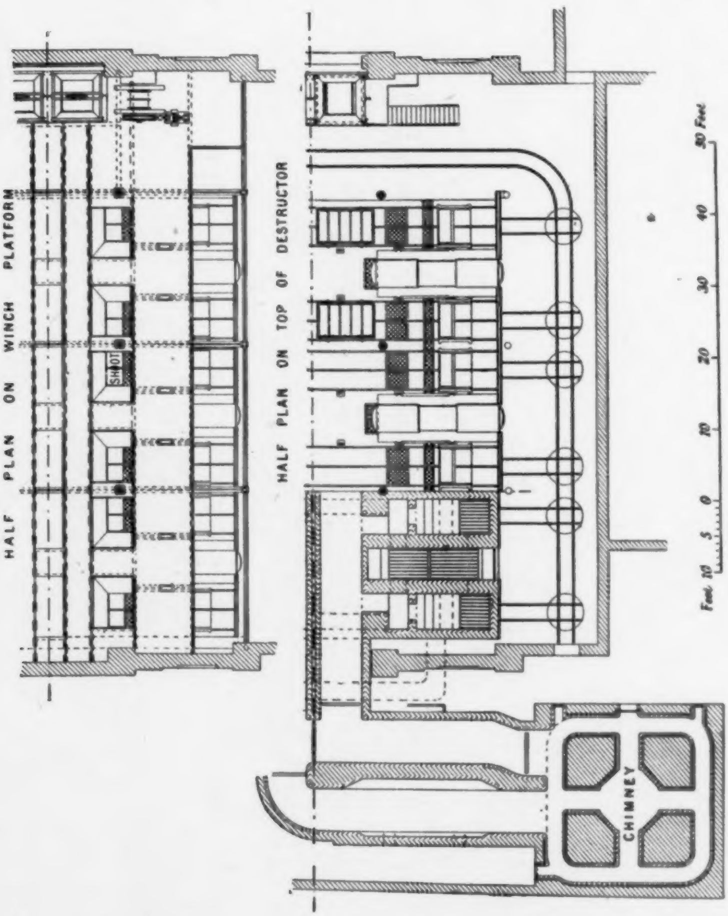


Fig. 478.—Section Plans.

mains. In practice things worked out very differently. First it was found that pumping cold water into the vessel into which high-pressure steam was being admitted caused such a water hammering that the practice had to be discontinued forthwith, and it was found necessary to partially heat the feed-water by passing it through a Green's economizer before putting it into the

thermal storage vessel; this, however, effectively got over the difficulty. Secondly, by the autumn of 1897 the demand for steam during the daytime had reached nearly the limit of heat it was possible to get from the refuse furnaces; so, having nothing to spare from that source, no additional thermal storage vessels were necessary.

18. The boiler-feed is now furnished by a Weir pump, which forces the cold water through a Green's economizer, where it is heated to a temperature varying with the load, although at light loads a maximum temperature of 250 degrees Fahr. has been reached. The feed then passes into the thermal storage vessel, Figs. 475 to 477, fixed at a level of about 20 feet above the boilers. This vessel is simply a cylindrical shell, 30 feet by 8 feet, which is used for storing, during the hours of light load, hot water with which the boilers are fed directly by gravity.

19. Most of the lime in feed-water comes down in the feed-storage vessel; the amount taken out of the vessel after a run of seven months was little short of one ton after being dried. The deposit in the economizer tubes was less than 1-16 inch in thickness; it was of a harder nature than that in the feed-storage vessel, and could be removed by a scraper. The boiler-tubes were frequently examined and were found fairly clean, the deposit in the tubes amounting to an average of 3-32 inch.

20. This system of feed-storage has undoubtedly contributed considerably to the success of the plant generally; it enables the engineer in charge of the steam-raising plant to store hot feed-water during about eighteen hours out of the twenty-four, so that at the time of maximum load the vessel is about two-thirds full of feed-water at a pressure and temperature equal to that of the boilers.* Tests have been made on several occasions, when the demand for electricity has been within the range of the refuse furnace to supply the necessary heat, and when, of course, no coal was used. The results showed 0.95 pound of water evaporated at a steam pressure averaging 130 pounds for 1 pound of refuse burned. A considerable reduction must, however, be made in these figures when taken over say twelve months. Damp weather

* For full details of Thermal Storage System as applied to ordinary steam raising plants, see Paper on "Economy of Fuel in Electric Generating Stations," by H. McLaren, M. I. Mech. E., Proceedings, Institution of Mechanical Engineers, 29 July, 1903.

(which affects the quality of the refuse), low barometric pressure, choked flues, warped doors, the starting up of furnaces, etc., may easily bring down the average results over a lengthy period to 0.5 pound of water for 1 pound of refuse burned. The existing vessel has, however, proved most serviceable for the storage of hot feed-water, and as a means of removing the impurities from the feed-water before it reaches the boiler. The economizer was not erected until some time after the works were opened, as it was not anticipated that the temperature of flue-gases would warrant its insertion. However, after the plant had been running steadily for some time, and careful tests had been taken, it was found that the flue-gases at the base of the chimney had a maximum temperature of 700 degrees Fahr.

21. Adjoining the refuse destructor and electro-generating station are situated the public baths and wash-houses, which contain:—

1 swimming bath, 100 feet by 40 feet.
1 " " 75 feet by 34 feet.
76 slipper bath.
50 troughs for clothes washing.

22. The exhaust steam after leaving the engines (non-condensing) in the generating station is carried by a 16-inch pipe to the baths, Fig. 479, where it is put through heaters which supply all the hot water necessary for the whole of this large institution. Live steam is also supplied direct from boilers for clothes boiling, etc.

23. This scheme for heating the whole of the baths and wash-houses has proved eminently successful, and has excited a great amount of interest among municipal authorities. No charge whatever is made to the baths for any exhaust steam. A charge, however, of not quite £250 (or $1,250) per annum is made for the live steam. The arrangement is a very economical one for the Baths Department, and worthy of serious consideration in cases where exhaust steam is available. The Free Library, situated close by, is heated by exhaust steam from the feed-pumps.

24. The amount of refuse destroyed is between 25,000 and 26,000 tons per annum. The residue (or clinker) amounts to from 33 to 35 per cent. of the weight. The cost for labor for burning the refuse is very high, compared with that at other destructor plants, and is one of the most serious factors for consideration. The nearer a destructor is to the center of a large city, the greater

will be the wage bill, and this fact must be borne in mind in considering any new scheme:—

	Per ton. s.	d.
Cost for handling and burning refuse, including yard men—average	2	6
Clerks and establishment charges	0	4½
Repairs and maintenance of cells and plant and cost of engineering stores	0	10
Total	3	8½

25. The average number of men employed in actually handling the refuse (per shift of 8 hours):—

Furnace men	4	
Top men	3	
Lift men	1	daytime only.
Yard men (laborers)	2	daytime only.
Foreman in charge of shift	1	

The above figures are for the fifth year's working, when considerable repairs were necessary to the furnace, linings, doors, etc.

The works are run on the eight-hour-shift system, seven days per week.

26. The amount of electric energy consumed in the burning and handling of 25,000 tons of refuse is as follows:—

	Units per tons.
Electric fans	4.0
" lifts and trucks	0.5
" lighting	0.48
Total	4.98

The cost of the destructor, including furnaces, flues, buildings, portion of cost of chimney, constructional ironwork, etc	£15,000 or $75,000
The cost of chimney was £2,790, two-thirds of which was charged to the destructor and one-third to the electric generating department	$13,950

The residue from London refuse consists of approximately—

	Per cent. of refuse.
Common clinker	30
Fine ash	3
Flue dust	0.5
Old tins and iron	0.5
Total	34

27. The disposal of clinker when produced at works situated in the center of large cities is a difficult and expensive matter—it is usually found impracticable to use more than a small portion for slab and mortar making. The greater portion has to be removed in vans to the outskirts of the city. This in the case of Shoreditch costs 2s. or 50 cents per ton of clinker, and should be taken

into account in the total cost for disposal. When building operations, road-making, etc., are going on extensively in the neighborhood the clinker from refuse destructors is of considerable value. When properly treated it may be made into most excellent paving slabs, bricks, concrete and mortar.

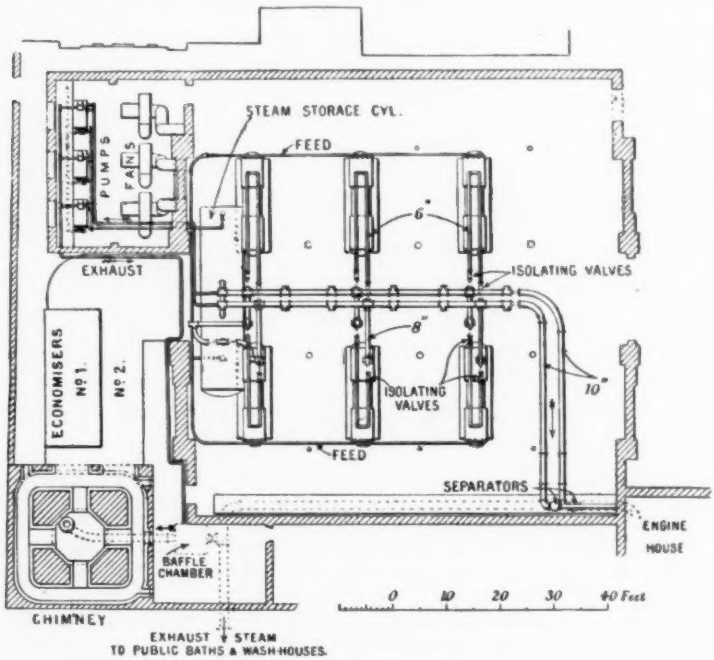


FIG. 479.—PLAN SHOWING PIPES.
Steam-pipes made of Steel. Boiler-Feed pipes, of Cast Iron. Steam-pipes to Pumps, of Copper.

28. No paper on refuse destructors would be complete without reference to the utilization of clinker for slab-making, etc., and the author gratefully acknowledges his indebtedness to Mr. E. J. Lovegrove, Borough Engineer to the Horsey Borough Council, London, for the results of his experience, extending over a number of years. Mr. Lovegrove says that rough clinker forms one of the finest classes of hardcore, as it provided a drainage bed so that the surface of the road was kept drier than was the case with other classes of foundation. As a concrete, mixed in proportion of one of cement to five of clinker (care being taken to have a sufficient amount of fine ash), it was superior to what was generally considered in London the best class of concrete, viz., that composed

of one part of Portland cement to five parts of Thames ballast; it was tougher and less subject to cleavage. This, on soils such as clay, where the ground was subject to expansion and contraction and the buildings to settlement, was a very important factor, which should not be overlooked. The same remark applied to the use of the ground clinker as a motor. Mr. Lovegrove had also used the material for plastering and rendering to a somewhat large extent; but, while it made a very hard plaster, there was one defect which showed itself to a greater or less extent, namely, that after the plastering or rendering had been finished for a few months, signs of blowing had occurred, causing small circular flaws about $\frac{1}{4}$ to $\frac{1}{2}$ inch in diameter in the face of the plaster; on examining these they appeared to be due to minute particles of iron in the clinker. In cases where this occurred the small places were cut out and made good, and no further trouble was experienced. At the same time, he considered it advisable to call attention to this risk, particularly when the rendering was used for water-tight tanks.

29. Very successful tar paving could be made by crushing the clinker and sifting it to the various sizes required for the bottom and top dressing, and this could be laid at a cost varying from 1s. 6d. per square yard to 1s. 9d. per square yard, according to the distance of cartage from the works. It appeared to equal in durability the ordinary granite tar paving, but it was not suitable for paths where the foot traffic was heavy. The use of the clinker for making paving slabs was not new. Many of the earlier hand-made slabs were faced with granite, thus losing one of the great advantages of the clinker slab, namely, the non-slipperiness of its surface. The adoption of the hydaulic press had done away with the necessity for granite facing. These presses had been in use for some time, particularly at Liverpool and Bootle, but at present the only machine of the kind in use in or near London was that belonging to the Hornsey Council. The slab produced was even in texture and non-slippery in wear, the effect of heavy foot-traffic, after four to five years, being practically imperceptible, so that it compared very favorably with other manufactured paving slabs, and the cost of production was considerably less than one-half the cost of ordinary granite concrete paving. The clinker drawn from the refuse furnaces was ground in a mortar-mill driven by steam obtained from the destructor boilers, and, after being sifted through a $\frac{1}{4}$-inch square mesh sieve, was mixed with Portland cement in the proportion of two of ground clinker to one of

cement. The concrete was then placed in strong iron travelling moulds and run by hydraulic power under the hydraulic press, which exerted a pressure gradually increasing up to $1\frac{3}{4}$ tons per square inch. The slab remained under pressure for about three minutes, when it was removed from the mould and placed on a rack to dry under cover for about a week, and was then stacked in the open until required for use, it being found that after three months the slab was in good condition for laying. The dimensions of the moulds in use were 3 feet by 2 feet, 2 feet 6 inches by 2 feet, and 2 feet by 2 feet, and the thickness of the paving was $2\frac{1}{2}$ inches. This process could not be looked upon in a large way as a method of disposal, but rather as a useful and economical method of using the wasteful material. The cost of manufacture was largely affected by carefully planning the works, so that there was sufficient mill-power to keep the presses at work, and the position of the mills being between the furnaces and the machinery room, unnecessary labor in wheeling was thereby avoided. He had carried out various comparative tests for personal information to ascertain the value of clinker-paving and briquettes, compared with other materials. With regard to these tests, it was not suggested that the most reliable test for paving could be obtained otherwise than in actual wear, whether of macadam for roads or paving for footways bore a very close relationship to the results obtained by the test. This test was applied by placing samples in a cylindrical testing machine, the cylinders being cast-iron and $11\frac{1}{2}$ inches in internal diameter, with three 1-inch by 1-inch angle-bar ribs riveted lengthwise on the inside of each cylinder; each class of material was subjected to 8,000 revolutions at a speed of 20 revolutions per minute. In the wet tests the materials were weighed dry, and about half a gallon of water was put in the cylinder, the weight after the test not being taken until the samples were again thoroughly dry.

30. As bearing upon the scheming and working of new plants the author makes the following suggestions, which are the outcome of practical experience from a Municipal Engineer's point of view:

1. The site should, when possible, be right away from residential and important business quarters and near to a tram route and a waterway.

2. It is unnecessary for the destructor to adjoin an electric generating station. It should, indeed, be far enough away or otherwise so placed that dust from works and clinker yard cannot reach the engine-room of the generating station.

3. Furnacemen should be made to rely on refuse as fuel; coal or coke should never be allowed inside a destructor works.

4. About 8 to 10 tons of refuse have to be handled to produce the same amount of steam as can be got by the combustion of 1 ton of good coal.

5. The use of lifts and trucks for handling refuse should be avoided where possible. Unnecessary handling of refuse soon runs up the costs. An ideal system from both a sanitary and economical point of view would be to shoot the refuse straight out of the collecting van into the furnace.

6. Fine dust is emitted from the chimney and will prove a nuisance unless flues are well designed and cleaned right out at least once a week.

The flue dust is of a gray color, very gritty, and thoroughly calcined; it is suitable for use as a basis for disinfectant powder, and large quantities are utilized in this way. When the dust is in a dry condition it will absorb anything up to 30 per cent. of its weight of pure carbolic acid.

7. Experience at Shoreditch shows that, in burning coal fires in conjunction with refuse, the full efficiency of the coal is not obtained as in an independent boiler, the reason being that, during the process of clinkering the refuse furnaces, a considerable amount of cold air unavoidably finds its way to the boiler-tubes. At times, also, when there is a scarcity of refuse and the furnaces are not working, as, for intance, on Sundays, it is impossible to obtain a high efficiency per pound of coal burned, owing to the impracticability of preventing cold air from finding its way through the dead refuse furnaces to the coal-grate. This drawback would appear to be one of the necessary evils of any boiler arranged to be fired by either coal or refuse. The trouble, however, could be considerably reduced, if not entirely eliminated, by an improved arrangement of dampers so fixed as to cut off entirely all possible admission of cold air from refuse-furnace doors.

8. By increasing the air pressure in the ash-pit to $2\frac{1}{2}$ inches it is found possible to burn 25 per cent. more refuse than with a pressure of only 1 inch; a harder and better class of clinker is also produced. Even when the air pressure is raised to 3 inches there is a considerable amount of combustible matter remaining in the clinker. In fact, from experiments carried out on a comparatively small scale under the author's superintendence for the Shoreditch Borough Council, it was found that, when ordinary unsorted

refuse was placed in an iron melting furnace (or cupola) heated in the first instance by a coke fire, the refuse contained sufficient burnable matter to melt itself down, and that this process could be carried on continuously for some hours without the addition of any other fuel. The residue was tapped in the ordinary way and came away in the form of molten slag of a vitrite nature and black in color. By this means the residue may be reduced to about 15 per cent. instead of 25 per cent. to 35 per cent. The slag may be run into moulds, and when cold is of a brittle nature. Mr. C. Wægner, of Berlin, Germany, some time ago obtained similar results; it was found in that case, however, that a considerable addition of coal or other fuel was necessary to smelt the refuse. The vitreous residue from smelted refuse, when pulverized, is suitable for the manufacture of glass paper such as is used by cabinet makers.

DISCUSSION.

*Mr. Charles Wicksteed.**—I have had no experience in dust destructors except this, that in the little town where I live, Kettering, we have recently had a dust destructor put down, and I as a member of the community took a great deal of interest in it. There was, I may say, violent opposition in the town against this refuse destructor being put in our midst, because it was thought it would be a source of considerable nuisance. And since I am on my feet I may say that, although the two papers presented to this meeting show the best thing we have been able to do in England so far, we are only beginners; we have a very great deal to learn. There is no doubt that these destructors are a nuisance, and it is impossible to conduct them by our present means without their being a nuisance. In the first place, the refuse is a lot of filthy muck and cannot be concentrated in one place without some sort of obnoxious smell, and, under certain atmospheric conditions, a great deal of smell, and a great deal of dust, too, not only in tipping it into the hoppers, but also in dealing with the clinkers, which are full of dust. The refuse from fishmongers and slaughter-houses takes a great deal of burning. They cannot burn it with the other stuff. They have to burn it at night. It must not only burn, but bake, or the clinkers will be offensive. The people in the immediate neighborhood of two

* Member of the Institution of Mechanical Engineers.

destructors I went to see all told me that they could not keep their windows open at night when the wind was blowing from the destructor.

Of course, in a little country like England, with a population about half what you have in this country, these questions naturally arise sooner than they do here. They arise in their intensity. I believe in the principle of destroying refuse by burning it; but we have a great deal yet to learn, and you must only take these refuse destructors as the best we have so far been able to put forth. We hope to do a great deal better not only in the way of burning, but in the way of abating the dust nuisance, which now we cannot prevent, and also to make the job a less objectionable one to the men who have to work at it. I never felt so dirty in my life as I did when I went around those refuse works and saw the poor fellows shoveling in the piles of refuse and perpetually poking the fires, which is necessary because one-third of the refuse formed into clinkers.

I hope this problem will not be too much for the resources of the American engineers, and I hope, too, that some day not very far distant, when we come again to visit Chicago, we shall be able to present to you a far better scheme or plan for a refuse destroyer than anything we have yet presented.

*Mr. Lewis A. Smart.**—On page 997 the author states that hydraulic presses "had been in use for some time, particularly at Liverpool and Bootle, but at present the only machine of the kind in use, in or near London, was that belonging to the Hornsey Council."

The Queens Engineering Works, of Water Lane, Leeds, make a specialty of such plants and have several at work in the neighborhood of London. The undernoted particulars regarding one of their plants which has been erected for about two years for the Woolwich Corporation may in whole, or in part, be of interest to members.

The clinker from the destructors is delivered into a yard where it is allowed to cool. When cold, it is ground in a 9-foot perforated pan mill of the edge-runner type. The ground material which all goes through $\frac{1}{8}$-inch mesh is then fed into a measuring machine where cement (about 20 per cent.) is added. The two ingredients fall into a differential mixer where the water is added. A further mixing and grinding is done by a 7-foot solid bottom,

* Member of the Institution of Mechanical Engineers.

edge runner, self-delivery, grinding mill, where the ingredients and water are ground and most intimately mixed together.

The material is delivered into the mould boxes of the flag press, where the flags are pressed and delivered ready for stacking.

Allied with this flag-making plant is a plant for making bricks out of the balance of destructor refuse available, and I append a few particulars of this. I have not the manufacturing costs of the Woolwich plant beside me, but similar plants erected elsewhere by the Queens Engineering Works are making excellent bricks which have a crushing strain about twice that of London stock bricks, and are being produced at 15s. to 16s. per thousand.

With regard to the brickmaking plant, the same preparing machinery is used. Instead, however, of using cement, about 7 per cent. of lime is necessary. The 7-foot pan delivers the material into a No. 1 "Hercules" brickmaking and pressing machine.

This machine moulds, presses and automatically delivers the bricks ready for stacking on platform wagons. Each platform wagon holds about 800 bricks, and the wagons, when full, are placed in cylindrical hardening chambers, where they are subjected to steam pressure at 100 to 120 pounds per square inch, for eight to ten hours, after which they are ready for use; that is, in about 12 hours from leaving the destructor.

The machinery for both the foregoing plants is all electrically driven, and otherwise of a thoroughly up-to-date labor-saving design. The Woolwich Council are using these bricks for most of their buildings, and, as already stated, they withstand rather more than double the crushing strain of ordinary London stock bricks.

*Mr. G. R. Dunell.**—I had not intended to speak, but some remarks made by Mr. Charles Wicksteed have prompted me to say something on the subject. From what Mr. Wicksteed said I fear gentlemen in America who are not acquainted with this subject will imagine that the destruction of refuse by cremation has been a failure in England. I differ from Mr. Charles Wicksteed, if that is his view. I have spoken to Mr. Wicksteed since, and I find that his experience has been with a class of destructors different from that with which I have had mostly under my own observation. It is the Horsfall destructor with which I am prin-

---

* Member of the Institution of Mechanical Engineers.

cipally acquainted, that is the invention with which Mr. Watson, the author of the first paper, is connected. It was my duty to make observations of this destructor, and I found as a result that it is possible to burn very green refuse without the slightest perceptible odor. In order to make perfectly sure that this is effected under the most adverse circumstances there is supplied a very large chamber lined with a refractory material. Shortly after the furnace has been in operation, this chamber is brought up to a white heat, and as there is provision made for an ample supply of air, perfect combustion is secured. I may also say that in this combustion chamber, a few days before I made my experiments, a dead horse was put in and was burned without offence. That does away with the obnoxious gas question.

We next come to the dust problem, which perhaps is more diffito deal with. Theoretically, of course, there is no difficulty. It is only necessary to reduce the velocity of the gases and give time for the dust to settle. Any dust is heavier than gases or air, and is deposited in a proper settling chamber. The vortex chamber is rather a short cut for that. I think Mr. Charles Wicksteed's experience must have been of rather an ancient date, or else with an unfortunate class of destructor. At any rate, it is negative testimony so far as concerns the non-effectiveness of cremation. The testimony I give is positive testimony. The thing has been done, and it can be done again.

Of course, destructors are like all other engineering structures, in that they should be designed on good principles. In fact, the designer needs to be very much of a scientific man. He has got to make arrangement for the proper admixture of air to complete combustion, and various other points which would be too numerous to go into here. In America if you want in your towns a perfectly healthy system that will do away with disease to a great extent, you must have destruction of refuse.

*Mr. C. Newton Russell.**—After reading Mr. Charles Wicksteed's remarks I can only assume that his experience of refuse destructors has been particularly unfortunate.

A great number of the refuse destructors in England are situated amid densely populated districts, and if considered or proved to be a nuisance would not be allowed by the authorities to continue working.

---

* Author's closure, under the Rules.

In the handling of refuse a large proportion of dust is unavoidable.

The dust trouble, however, is only experienced after the refuse has been burnt, and being well calcined *is inocuous.*

With the experience gained during the past twenty years, and with a well-managed modern destructor plant, the nuisances mentioned by Mr. Wicksteed should not exist to a troublesome extent.

A great deal more care and skill is in my opinion required in the proper burning of refuse than the man in the street dreams of.

It is quite possible to burn green refuse, offal, etc., without the slightest smell being made, providing it is fed into hot fires in comparatively small quantities. All these points have to be watched and arranged by the Chargeman.

Mr. Dunell's remarks do not call for any reply on my part.

Generally speaking, I am in agreement with him, but cannot quite swallow his story about the dead horse.

The remarks of Mr. Lewis A. Smart remind me that the Bermondsey Borough Council (London) have also a hydraulic flag-making machine in operation at their refuse destructor works, and it is, I understand, working very satisfactorily.

Some little time ago I had an opportunity of inspecting samples of bricks made from clinker and lime by a similar process to that described by Mr. Smart, and am in entire agreement with the results obtained. The bricks so made are much superior to the London stock type, and absorb considerably less moisture.

Mr. W. F. Goodrich * wrote that at the present time the American engineer was looking to the old country for guidance in successfully tackling the problem of sanitary refuse disposal, and to such as have closely followed recent British practice, Mr. Russell's paper must be very bewildering. The author concluded the paper by offering some suggestions, the first of which was as follows: "The site should when possible be right away from residential and important business quarters. . . . It is not necessary for the destructor to adjoin an electric generating station. It should indeed be far away or otherwise so placed that dust from the works and the clinker yard cannot reach the engine-room of the generating station." The writer could not agree with this

* **Added after the meeting.**

conclusion; a destructor should be located in that position where it would be of the greatest benefit to the citizen, firstly bearing in mind the fact that the generating station was usually erected on a central site, thus reducing the cartage cost to the minimum; and secondly, the facilities there presented for the utilization of the available power. It had been conclusively proved that a destructor might be erected and operated in a central position with a complete immunity from nuisance. He could not agree with Mr. Russell as to the relative positions of the destructor and the generating station; the nearer the steam boilers in the destructor-house were to the engines in the power-house the better. It was a simple matter in the design of the buildings to prevent dust from the destructor building reaching the engine-room; such a trouble had not been heard of in connection with recent practice. The author should have pointed out that, owing to the mechanical handling, lifting and tipping of the refuse at Shoreditch, an altogether unusual amount of dust was liberated.

The author observed that "the nearer a destructor is to the center of a large city, the greater will be the labor cost." This was an extraordinary statement, and moreover very illogical. The labor cost in connection with the operation of any destructor must in the main be determined by two factors: (*a*) The rate of wages ruling in the city and (*b*) the method employed for the handling or charging of the refuse into the destructor. It was scarcely necessary to add that the rate of wages ruling in a city would be the same whether paid at a destructor works in the center of the city or two miles from the center. The real determining factor was the design, general arrangement and method of handling the refuse; if these were wrong the geographical location of the destructor could not set them right. The author further remarked that: "The use of lifts and trucks for handling refuse should be avoided where possible; unnecessary handling of refuse soon runs up the costs." In these few words they had the complete explanation of high labor cost, the central location of the destructor or otherwise being beside the question entirely.

In the paper entitled "Combined Refuse Destructors and Power Plants," read before the Institution of Civil Engineers in December, 1899, by Mr. C. Newton Russell, the following conclusion was presented: "That a combination of one boiler and two furnaces, arranged as shown in Fig. 477, may be relied upon to evaporate 2,888 pounds of water per hour from and at a temperature

of 212 degrees Fahr., to a pressure of 200 pounds with refuse as fuel." Recently at Chicago the same author explained that this same arrangement of destructor cells and boilers was inefficient; he also observed that the earlier figures did not represent average results which were approximately only half as good as those previously quoted as being reliable. The writer was in complete agreement with Mr. Russell when he stated "that coal or coke should never be allowed inside a destructor works." Several years since this opinion was held and clearly expressed by many engineers. The arrangement of cells and boilers, which Mr. Russell eulogized in 1899 and condemned in Chicago in 1904, however involved the use of coal or coke in the destructor-house; it was only in this way that the boilers which were arranged for supplementary coal firing could be so utilized. In the paper of 1899 he also clearly stated as a conclusion that: "Domestic refuse in London has an average calorific value equal to 0.99 pound of water per pound of refuse burnt." Now, in the present paper this figure was practically divided by two, an average result being given of 0.5 pound of water per pound of refuse destroyed.

At Chicago Mr. Russell further observed that: "All successful destructors have furnaces operating on practically the same principle, the only difference being in details . . . similar results are being obtained from the different destructors erected by different makers, each claiming superiority." Now, there was no evidence of any kind submitted in the paper to support this statement, and it must be clear to all that had Mr. Russell not derived his experience from one destructor installation alone he would not have thus written. There was a very wide difference in the principles of design and arrangement of the various British destructors; all makers professed to aim at and secure equally good results, but profession was not necessarily practice. The striking difference in design alone would not permit of equal results being obtained all round. Efficient combustion was governed by certain cardinal principles, which would not be questioned. To observe such principles was to command success; on the other hand, to ignore such principles involved inefficient working.

Although the author spoke hopefully of clinker utilization, he altogether failed to convey the extraordinary progress which had been made. They were told, for instance, that only one clinker-flag plant was in operation "in or near London"; this was incorrect. As a matter of fact, such plants were in operation for the

following metropolitan boroughs: Woolwich, Bermondsey, Fulham and Battersea, and on the outskirts at both Ealing and Walthamstow, in addition to Hornsey, which single installation was referred to by Mr. Russell. The metropolitan boroughs of Woolwich and Fulham also had complete brick-making plant in daily operation. In the provinces clinker flag plants had been adopted by the following municipalities: Liverpool, Leicester, Cheltenham, Birmingham, Bootle, Sheffield, Bradford, Bristol, Oldham, Blackburn, Withington, etc.

Mr. Russell touched but lightly upon the economic aspect of combined electricity and destructor works; some would say that there was no economic aspect, but whether or not, the electrical engineer often sought to obtain the available power gratis, or at a very low figure. In the combined works one might as a rule look for the minimum cartage cost of the refuse; the citizen therefore benefitted in this direction, as also by reason of the fact that the collection was accelerated. The citizen should benefit by the production of power, but naturally this was entirely governed by the efficiency of the destructor as a power producer and the price mutually agreed upon for the power. As he had already observed, Mr. Russell stated that all destructors were practically alike and equal in efficiency; this being so, then perhaps the author would inform them why such varying results in power production were recorded with destructors combined with electricity works in London? That there was a very remarkable difference was clearly shown in the following Table:

Metropolitan Boroughs.	Board of Trade Units generated per ton of refuse destroyed.
A. Shoreditch	20
B. Fulham	26.62
C. Hackney	54.19
D. Woolwich	57

A. = Average for one year, good day load.
B. = " " " " " "
C. = Non-condensing, excellent day load.
D. = " lighting load only.

These figures by no means represented the best results yet obtained in electrical output per ton of refuse destroyed. At Cleckheaton, with an average of 12 tons of refuse daily and a traction load, an electrical output of over 70 units per ton of refuse destroyed was frequently obtained; nor was this a solitary example. At Plumstead combined destructor and electricity works of the Metropolitan Borough of Woolwich, during a 24-hours' test con-

ducted by the National Boiler Insurance Co., an output of approximately 100 units per ton of refuse destroyed was obtained at the peak. This unique result was clearly set forth in the accompanying diagram (see Fig. 480).

Mr. C. Newton Russell wrote, in reply to Mr. Goodrich's communication, that there were a few points in the criticism calling for notice and explanation. With reference to the dust question, nearly all the inconvenience was caused by the hot clinker, which had to be tipped in an open yard and which lay there until cool before it could be removed. With even a light wind, the dust was liable to be blown about, and to become a nuisance, although

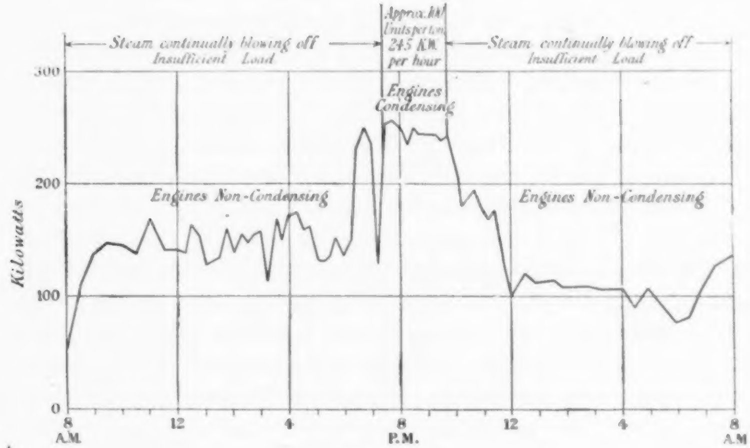


FIG. 480.

perhaps not doing any actual damage. The inconvenience caused by dust in the destructor house was insignificant compared with that from the open yards common to all refuse destructors seen by the author.

The author agreed with Mr. Goodrich that it was possible from a purely engineering point of view to so design works that dust would be entirely prevented from reaching the engine-room; but the increased cost of so doing would be, however, not worth the outlay, for it must be remembered that this paper dealt with destructors for large cities where the demand for electric (or other) power was so great that the proportion of steam-power that could be generated from the destructor furnaces was perhaps only one-twentieth part of the total required. Mr. Goodrich had evidently

lost sight of the difference in revenue earning capacities of coal-fired electric generating stations and destructor plants; in many cases it would pay commercially to put down the former plants, where it would (owing to the high price of land) be ruinous from a business point of view to install a refuse destructor simply to demonstrate that it could be done. The value of land in Shoreditch was such that the interest and repayment of capital represented an annual charge equal to one shilling per ton of refuse destroyed, or, in other words, sufficient to purchase the freehold of land in the outskirts of some towns in three or four years. The author maintained, moreover, that what energy it was possible to generate by a destructor situated away from the center of a town could easily be transformed into electrical energy by means of a small plant and fed into the nearest supply mains with the additional advantage of boosting up the electric pressure at a point where in all probability it would be badly wanted.

With regard to the different results obtained at various destructors, set forth in a table given by Mr. Goodrich, the author was glad to have the opportunity of explaining them, as the figures showed a difference of efficiency of more than 100 per cent., and thereby inferred that the same difference of efficiency existed between different designs of destructor plants, which was not a fact. The figures as set out by Mr. Goodrich were on a wrong basis, inasmuch as the efficiency of steam-generating plant had not been given. The figures were, therefore, misleading; for instance, the steam generating plant at Shoreditch consisted of a number of small generating sets connected to very long steam ranges and run non-condensing. This plant, installed in 1897, was now practically obsolete so far as efficiency was concerned, and used about 40 pounds of steam at least per unit generated. Mr. Goodrich compared this and the alternating generating plant at Fulham with two quite new stations where the steam plant was more modern and condensing, and where, at least in the case of Hackney, a kilowatt per hour could be produced with an expenditure of about 20 pounds of steam.

The figures given in Mr. Goodrich's table, when put on a proper basis, therefore, only went to prove the author's contention that the differences in efficiencies of destructors was not so great as Mr. Goodrich would make out. The author did not hold any brief for any destructor-makers, nor did he wish to hold up the Shoreditch destructor as a paragon by any means; moreover, he was

quite ready to concede that the destructors with which Mr. Goodrich had had experience were perhaps the most efficient, but he strongly objected to efficiencies of plants being compared in terms of units generated, unless the full story was told, and the number of pounds of steam it took to generate a unit of electricity was given in a parallel column for each case cited.

With this explanation, the author felt sure that Mr. Goodrich would agree with him that the figures given, that is A.B.C.D., were of little value for comparative purposes.

No. 1045.*

THE POWER PLANT OF TALL OFFICE BUILDINGS.

BY REGINALD PELHAM BOLTON, NEW YORK, N. Y.
(Member of the Society.)

1. New York City may be considered as being legitimately the home of the tall office building, since the restricted area of its business center has by nature been limited in all directions but one, and affords a real justification for the increase of its area by vertical extension.

2. Moreover, the character of the material forming a foundation for its extremely tall structures is admirably adapted to the imposition of these extraordinary heights, and tends to make the problem of their support comparatively simple. With the exception of the bed of glacial sand laying between the two ridges of the rock of lower Manhattan, which in itself has formed a not undesirable support of a permanent character, the footings of tall buildings are carried on solid rock, in places being extended by caissons through the bed of glacial débris.

3. Secured thus by nature from doubt as to the permanency of their support, the limitations of height become only those imposed by practical considerations above ground, and the present practice has settled down to a general adoption of a height affording space for 16 to 20 stories, the extreme presented by the Park Row (27) and the St. Paul (25) not having justified a general imitation.

4. In point of fact, the limitation of height has come about in a natural manner by realization of the disadvantages of the remoteness of such lofty floors from the street, and the excessive expense of maintaining a suitable elevator travel schedule on such long runs, the burden of which excess is to be charged to the

* Presented at the Chicago meeting (May and June, 1904) of the American Society of Mechanical Engineers, and forming part of Volume XXV. of the *Transactions*.

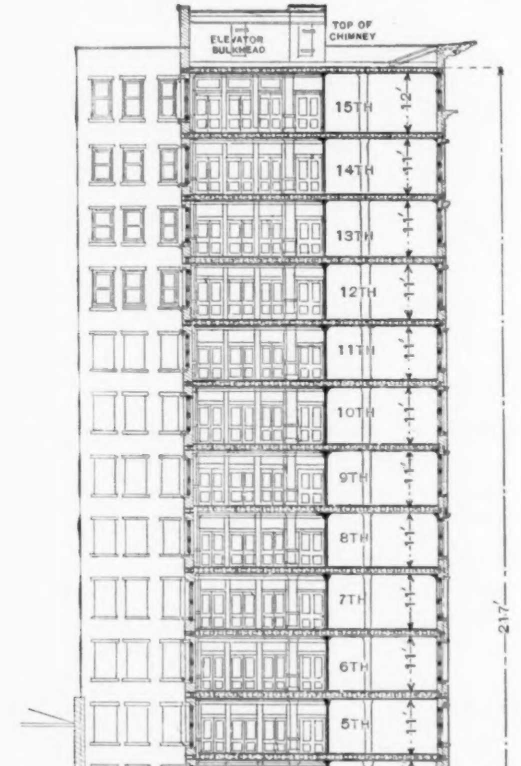


FIG. 481.—SECTIONAL ELEVATION OF A TYPICAL COMMERCIAL SKY-SCRAPER.

upper floors, and the returns from which are often insufficient to make the expense commercially successful.

5. The provision of express elevators, making stops only above a certain level, is found necessary in buildings exceeding 16 stories, and in some of that height also, and is usually made to serve from the ninth or tenth level, or, roughly speaking, the upper half of such structures. The addition of floors in excess of 20, would render it necessary to raise the level of the express stop, and would place certain of the intermediate floors at a disadvantage. The addition of more elevators is thus the only proper provision for such added floors, and the burden of their maintenance, together with the increase of travel of the whole of the rest of the bank of machines, falls upon the added floors, which thus become of an increasingly expensive character.

6. In some buildings, such as the Empire (20) and St. Paul (25), certain of the elevators are operated as locals, operating only to a certain floor. Owing, however, to lack of foresight and provision for this arrangement, these machines have been constructed the full height of the building, occupying space uselessly, and having the travel of pistons in their cylinders limited to the lower portions, and of their ropes to a part of their length. Such machines, even if eventually their travel should be extended, would probably require reboring of cylinders.

7. The proportioning of the number and travel of elevators to the area, character, and height of these buildings has, so far, proceeded in a very haphazard manner, mostly by mere imitation of others, and often without any allowance for increase of stories, and attempts have been made by architects responsible for these blunders to mitigate the disadvantages thus imposed on the buildings, by increasing the floor area of the cars, and by increasing the speeds of the machines. Both are subject to practical limitations, mere increase of area of cars being the cause of delay in handling the passengers, and the speed of travel being practically limited to 600 feet per minute for stopping or way cars, and to 750 feet for express cars, beyond which speeds more time is lost than gained by over-running the landings.

8. In an article on this subject (Cassier's Magazine, volume xxi., No. 3, January, 1902) the author dealt more fully with this problem, and pointed out that a basis for the due proportioning of elevator service may be found in the provision of an elevator to an area of 1,000 square feet of rented space repeated on 16

floors. For heights exceeding 16, the basis of calculation should be the above, plus a provision for an average number of opportunities of access to cars at the average or medium travelling floors. Special conditions exist in certain buildings tending to increase travel at certain periods, and these are of course difficult to ascertain in advance of occupancy. As an example, may be mentioned the German-American building, 15 floors, on the thirteenth floor of which is a largely attended law-school, the students in which emerge and arrive at stated hours, and throw the burden of several hundred impatient travellers upon the car service.

9. The maintenance of a satisfactory service, and of its economical operation is more dependent upon the personal element than is commonly supposed. Passengers' habits have much effect. In New York, the general travelling public has learnt to make use of the signal system, to step promptly in and out of cars, and to announce their destination well in advance of arrival, all of which contribute much to the regularity and economy of schedule operation.

10. The importance of satisfactory signal systems between floors and cars, announcing the proximate arrival of cars to waiting passengers, and of the presence of the latter to the operators, has had a more important bearing on the economy of operation than any economic improvement in operative machinery. There still remains the uncertain and often unsatisfactory element of the personality of the car opeartor, who is only rarely made cognizant of the bearing which his promptitude, attention and care in avoiding over-running of landings has upon the cost of operation. Where these employes have been placed, as they should be, under the orders and directions of the operating engineer, remarkable improvements have resulted, particularly in the operation of electric elevators, in some of which the controls are of so small and delicate a character, that operators get into habits of continually fidgeting with the little levers.

11. The electric elevator is liable to peculiar misuse in this respect, and suffers more than the hydraulic system from the losses of over-running and of useless stops and starts by failing to make precise landings. Partly by these and partly by the inherent unsuitability of electric distribution of power for this purpose, the electric elevator has during ten years of use and improvement, failed to dislodge the hydraulic system as the most reliable and suitable method of schedule operation.

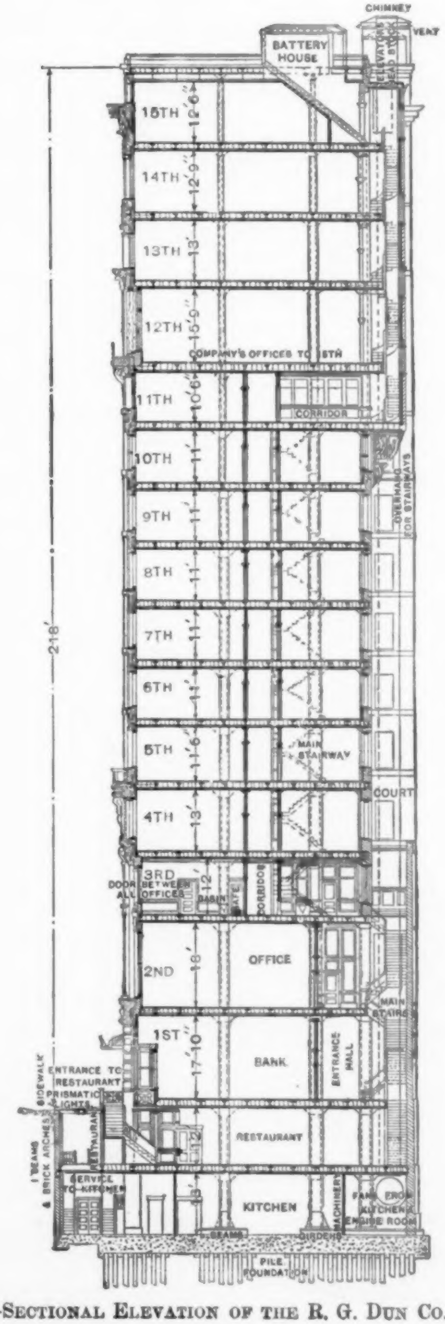


FIG. 482.—SECTIONAL ELEVATION OF THE R. G. DUN CO.'S BUILDING.

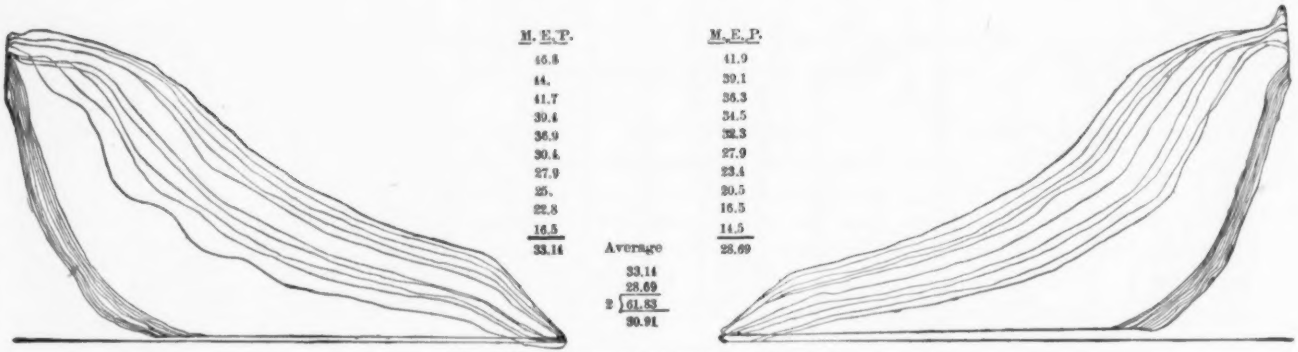


FIG. 483.—SAMPLE INDICATOR DIAGRAM.

Taken during Test at Lords Court Building 1.35 P. M., July 30, 1896. Ames Engine No. 14737—16" × 14"—246½ Revs. Scale of Spring = 50. Ten Cards at one Minute Intervals I. H. P. Average = 108. I. H. P. By Largest Card = 154. I. H. P. By Smaller Card = 54.

12. The peculiar advantage of electric force in the distribution of power over long distances, does not, in the case of elevator operation, fill any requirement, since the source and expenditure of power are close together.

13. The irregularities in car operation, even in schedule work, are severe, and without the inclusion of storage of force, throw severe strains upon operating machinery, with accompanying loss of economy due to wide variations of load. The following diagram of varying loads on a simple engine operating direct on Sprague screw electric elevators, will illustrate the severity of the work. The equivalent of the storage of the hydraulic system, which is effected by the provision of an air-pressure in closed tanks, is, in the electric system, required to be the extraordinary expense of a storage battery, and there are only a few buildings employing electric elevators on schedule service of any magnitude that have not found such an expensive addition necessary.

14. They are installed in the Commercial Cable building (21), Dun (16), German-American (15), Park Row (27), and other buildings.

In the Dun Building the battery stands for a capital cost of \$18,000, representing annual maintenance and interest of \$2,150, the equivalent of nearly 20 per cent. of the coal bill. In addition, one engineer is constantly employed solely to attend to the operating mechanism of the five elevator machines, a further burden of \$1,000 per annum.

15. The maintenance of the form of electric elevator known as the Sprague-Pratt, or screw machine, and to a certain degree of the other types of drum machines, is expensive, and out of all proportion to the cost of maintenance of an hydraulic mechanism. The former have been peculiarly subject to wear and tear. Screws, thrust plates and ball-bearings require renewal on an average every eighteen to twenty-four months. Constant attention and repair is required to the numerous details of the controlling mechanism.

16. The work of repair and replacement of parts of electric elevators affords profitable employment for an entire works in this city, and further construction of the screw type has been definitely abandoned by its owners.

17. The apparent economy of operation of an electrical system entirely disappears under analysis of these conditions, and the electric elevator stands to-day an economic failure for schedule

service, while in point of simplicity, safety, ease of maintenance and control, the hydraulic elevator has maintained its reputation.

18. It may be freely admitted, nevertheless, that there are cases where the adoption of electricty as the motive force for elevator operation, is desirable, such as when economy may be effected in other directions by their adoption. Thus, in buildings which can be more economically operated by the supply of current from the public service, the reduction of the labor bill may outweigh the additional cost of operation. In such service the operating supply of a schedule elevator will cost from $1,200 to $1,500 per annum. One such car was run on public supply, in a bank of five, 15 stories, the bill for current being $130 per month.

19. Electric machines of the drum type have now been operated to speeds as high as 600 feet, but are in this work usually applied where speeds of 450 feet suffice. An installation has recently been made, in the Arthur Building (18), of a flying rope-driven electric elevator, which has had the misfortune to be the cause of serious accident, due to failure of its limit-stop devices on the down run. The screw machine has in one instance suffered the breakage of its frame, due to an unequal strain caused by the pulling out of one cable from an eyelet. The lag in starting of an electric machine is also a source of latent danger. The operators, relying on this lag, do not close doors before opening the lever, and this contributed to a fatal accident.

20. New York architects still generally adhere to the proportion of cars, narrow across the entrance and often unduly and disproportionately deep.

21. The disposition of cars also seems to be a matter of wide variation. In the larger buildings, where sufficient space is available, chiefly those of banks and large companies, the arrangement in a semicircular form is desirable. Such are the Havemeyer (16), American Trust (23), St. Paul (25), the Corn Exchange Bank Building (21), 42 Broadway (22), Fig. 484, and the Park Row (27), Fig. 485. In the latter, however, cars are also placed beyond the semicircle at each side, and are thus at a disadvantage as to accessibility. In others, such as Commercial Cable (21), North American Trust (14), Queen (15), Fig. 486, Broadway Chambers (18), Fig. 487, and Kuhn-Loeb (18), the cars are placed in a line along a narrow corridor or hall. These suffer if the bank be too long, or if, as is the case in several, such as the

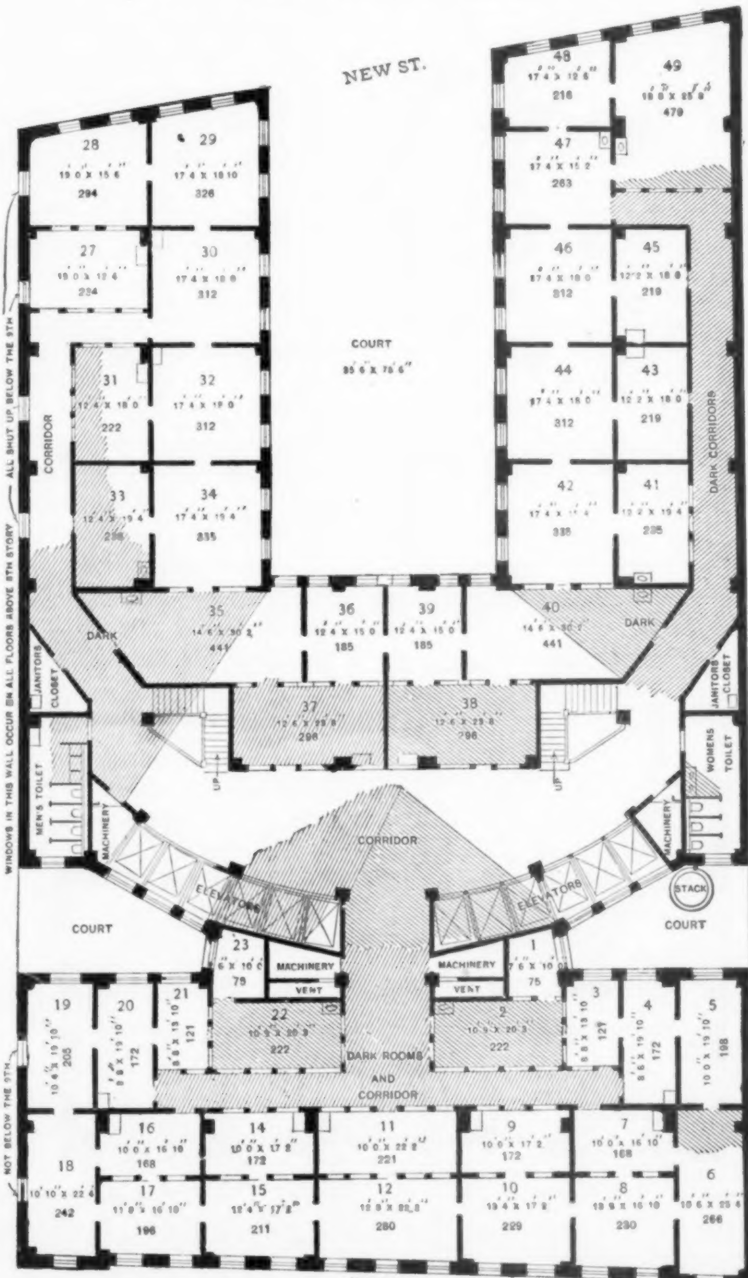


FIG. 484.—FORTY-TWO BROADWAY. TYPICAL FLOOR PLAN.

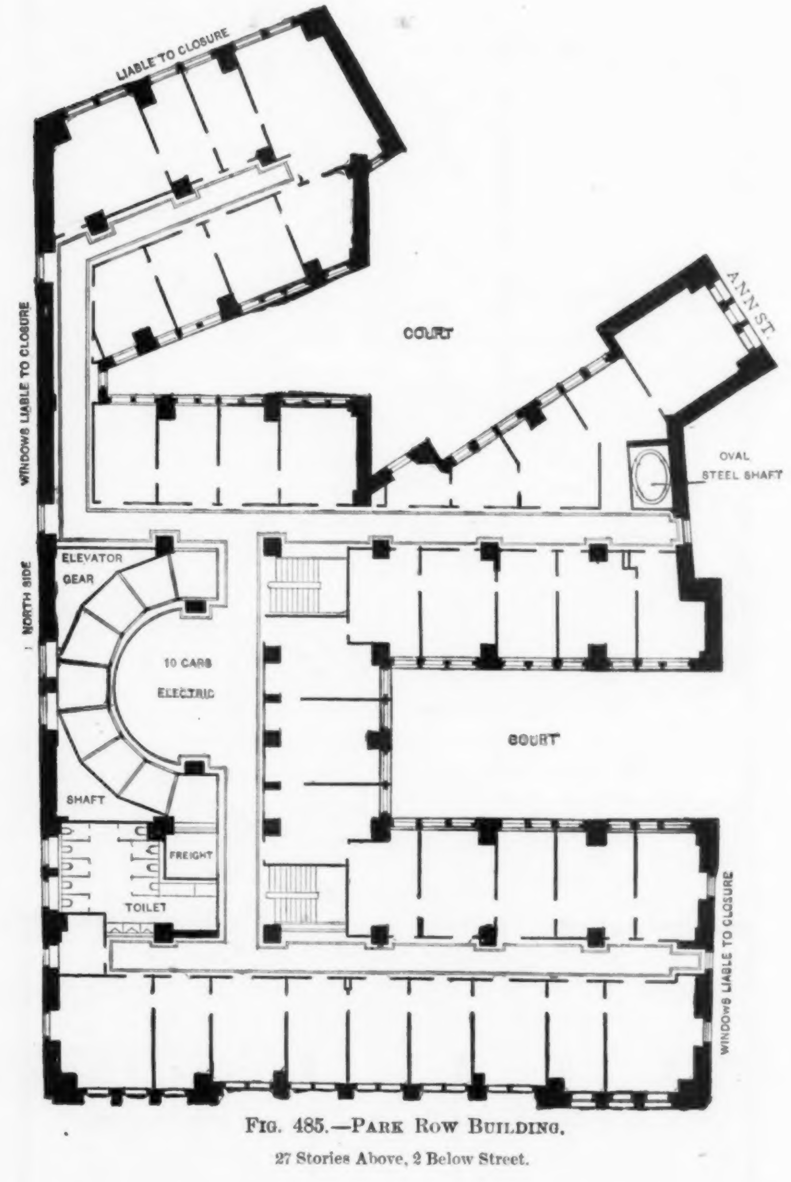


FIG. 485.—PARK ROW BUILDING.

27 Stories Above, 2 Below Street.

Empire (20), Bowling Green Offices (16), Fig. 488, North American Trust (14) and others, entrance to cars be placed between projecting pilasters. In other buildings the cars are placed in two banks facing each other, an arrangement not to be commended. Of such are the American Surety (21), Johnston (16), German-American (15), Fig. 489, and Hudson (16), Fig. 490.

A curious arrangement is that of the Central Bank Building (16), Fig. 491, where two cars are placed at right angles to three others, and the sixth machine at the end of a long corridor.

22. Before leaving the subject of elevators, the opportunity may be taken of again drawing attention to the danger which their open shafts offer in case of fire. In every instance of a fire in these buildings, the smoke has promptly found access to the elevators and rendered them useless. Such occurrences took place in the Times Building (12) and World Building (14). The shafts should be enclosed separately and the doors and grillage filled with wire glass or metal. The same consideration applies to the stairways.

23. The maintenance of elevator service during the business hours being of prime necessity, and of a very definite and regular character, the heat available in the form of the exhaust steam from the motive machinery has to be taken into account in connection with the warming of the building, a matter which is only of a slightly less degree of importance.

24. Other services, such as electric lighting, being irregular, and house water, being comparatively small in extent, the exhaust of elevator power is the main source of heat supply during the cold season. This exhaust in heavy schedule service is nearly always in excess of heat requirements during the business hours, except in certain very economical buildings, such as the Bowling Green Offices, where the whole exhaust is utilized in the work of heating.

25. The adoption of the Webster vacuum and Paul air suction systems, has, by reducing back-pressure on engines and pumps, reduced this excess. Certain buildings provided only with the ordinary one and two-pipe gravity systems, require a back-pressure in extreme weather of from 5 to as high as 15 pounds to effect complete circulation. The New York Life Building (13), American Tract Society Building (23), and Park Row (27) were all benefitted in this respect by the addition of vacuum systems to their heating apparatus.

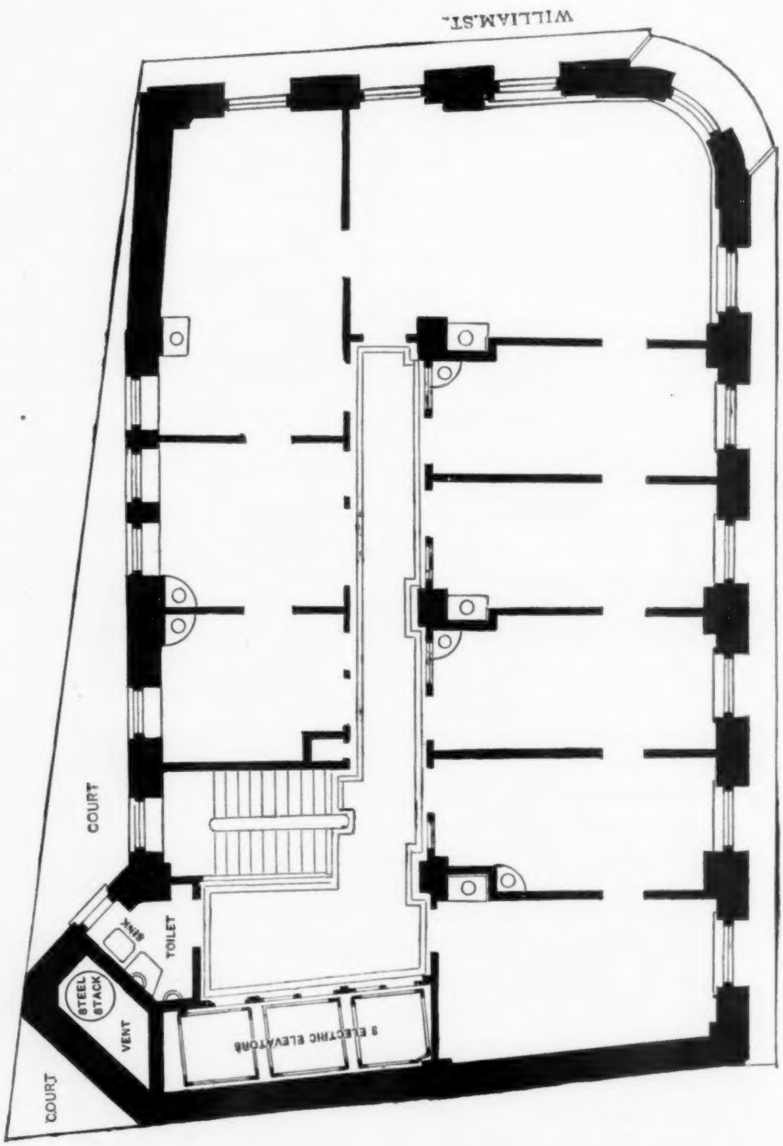


FIG. 486.—QUEEN INSURANCE BUILDING.

15 Stories Above, 2 Below Street.

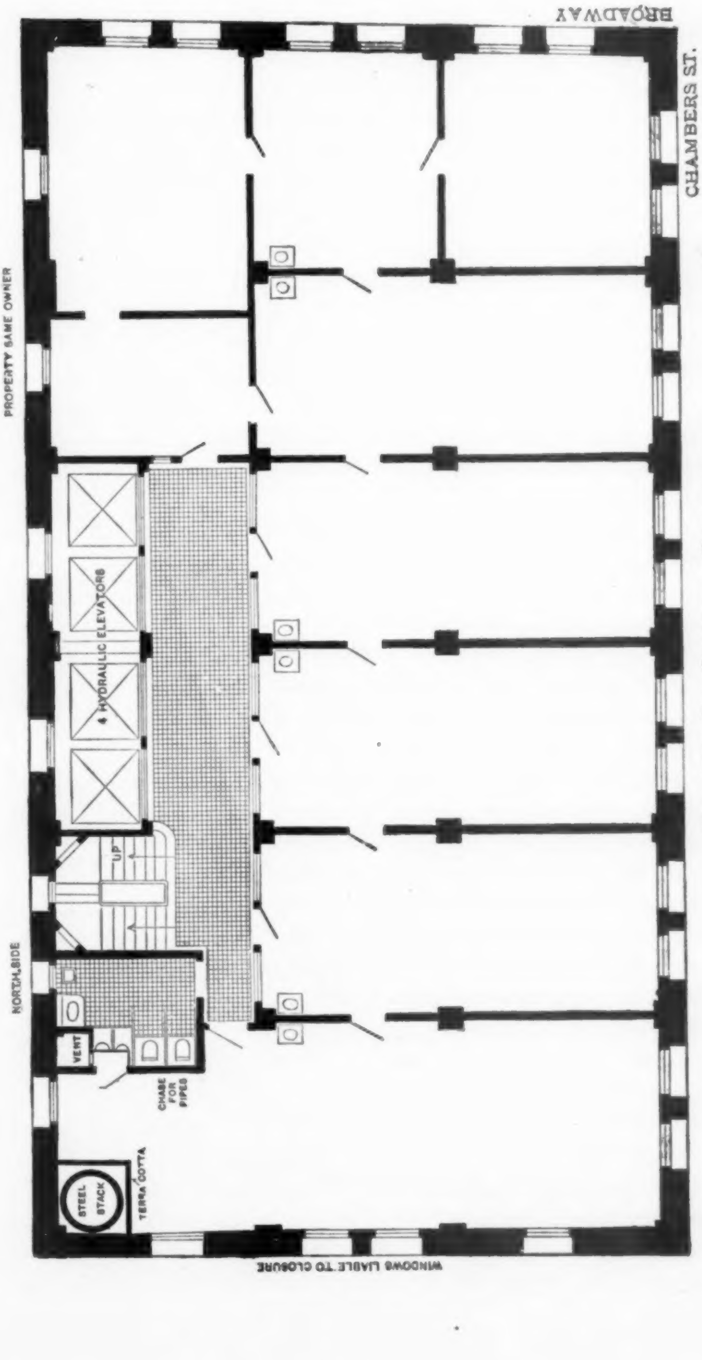


FIG. 487.—THE BROADWAY CHAMBERS.
18 Stories Above, 2 Below Street. Covers Entire Area.

26. The Webster system has been in use in the Lords Court (once 15 and now 19), Fig. 492, Bowling Green (16), Hudson (16), Central Bank (17), and in the German-American (15), in which it was for the first time installed in a tall office building, with entire success, for periods of five to eight years, circulating with a pressure not exceeding half a pound through headers and risers in some cases over 450 feet in length. The reduction in size of piping which can be effected by this system is a considerable advantage in cost and in the effective concealment of risers and branches.

27. The most desirable result with this system is the control of temperature in the radiators, by which the common complaint of over-heating of offices may be avoided. The ordinary two-pipe gravity system requires the relative adjustment of two valves to reduce heat, a nicety to which the average tenant is not willing to devote time.

28. The one-pipe gravity system practically involves full heat or none in the radiator, for which reason its adoption has in New York office buildings not met with much favor, except in the cheaper class of speculative buildings.

29. The overhead system of supply has been adopted only in a very few instances, such as the Broadway Chambers (18), owing to the unwillingness of owners and architects to provide a roof-space for the distribution of pipes.

30. Indirect heating of these buildings except in certain special spaces, such as bank parlors, has been practically abandoned, with the exception of the Hanover Bank Building (22). This recent building has also installed refrigerating apparatus, designed to reduce the temperature of the air in the corridors in warm weather, and a very large attempt in the same direction is to be made this summer in the New York Stock Exchange (8), with a refrigerating plant of 400 tons rated capacity. While the cost of such operations does not enter into account, compared with the convenience, and while the supply of exhaust steam in warm weather may bring that cost within reasonable consideration, the problem of water supply for the purpose of condensation is one that will probably prevent its present extension.

31. The present pressure in the Wall Street district is at noon barely 5 pounds per square inch, and the extra draft imposed by the large plant mentioned will probably bring about a worse state of affairs. The water supply being thus defective, all the tall

buildings are provided with receiving tanks and with pumping systems, by which the necessary pressure is raised for house supply. The storage is commonly limited to a few thousand gallons, so that a failure of water supply becomes a very serious matter in the down-town district in a very short time.

32. The water for house and fire services is either raised to roof tanks, or is put under equal or higher pressure on stand pipes or air-drums in the lower part of the building. The divided pressure system, installed by the author in a number of buildings, reduces the amount of pumping required, and has operated continuously and satisfactorily for nearly ten years.

33. Nearly all well-equipped buildings now utilize filters upon their whole supply, with beneficial effect on fixtures and feed, by the elimination of sand and mud. The feed system is in summer a direct cold feed from outside supply, the entire exhaust being wasted, except in those cases where an open heater is used, by which about 20 per cent. of the feed water is returned as condensed exhaust.

34. In winter, the feed is mainly composed of condensation from the returns of the heating system with a make-up by cold supply, which in the author's practice is heated by passage through coils in the drip-drum, where all oily and blow-off wastes are collected and cooled previous to discharge to the sewer. Such discharges, and usually all drainage of sub-basements or cellars, require in New York to be lifted to the sewer levels by pumps or steam syphons.

35. The presence of surface water in the rock seams and in the sand below such floors renders necessary special provision for water-proofing, especially around the ash-pits of the boilers. One method is to place the boilers in a steel plate pan extending below their entire footings. Another is to sink a sump pit or pits at suitable points, and lay courses of hollow tile drains to the pits from which the water is pumped.

36. In the Dun Building (15) the author found a sufficient supply to justify sinking a tube well 30 feet deep below cellar floor, whence a pump discharges the supply, which is fairly clear but chemically impure surface water, over the coils of the refrigerating plant, thence to the eleventh floor to a tank. From this tank the air supply of an Adams sewage lift is compressed, and by it all sewage of the kitchen, of the restaurant as well as that of the basement toilets, is lifted to the sewer. The water,

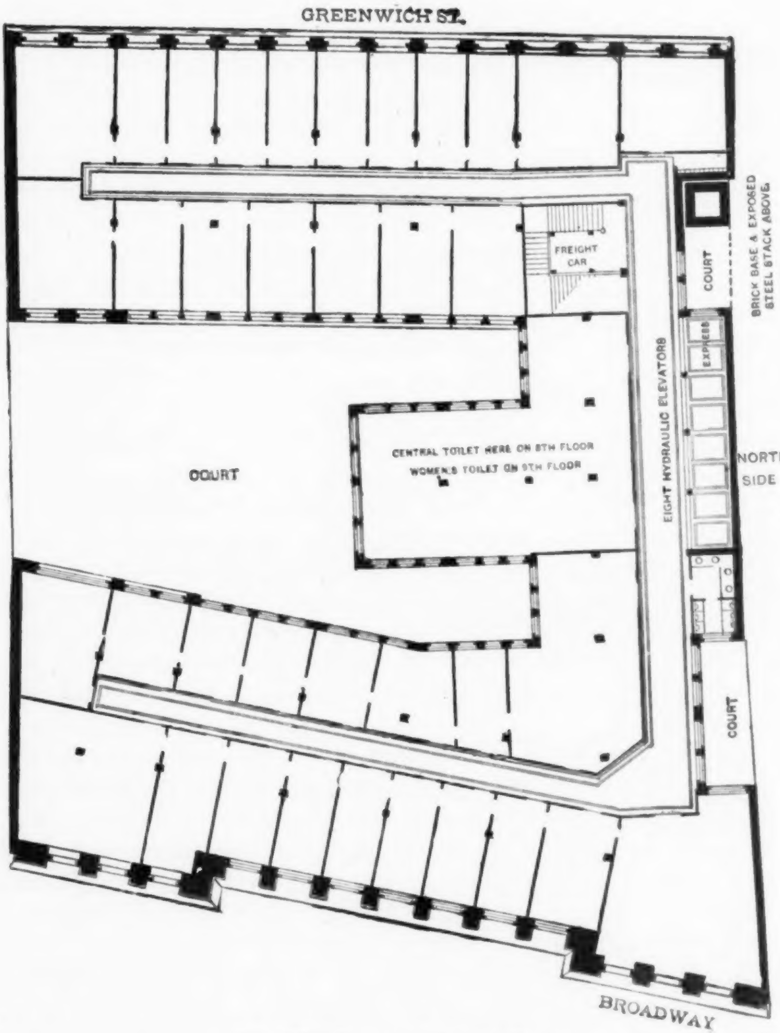


FIG. 488.—BOWLING GREEN OFFICES.

16 Stories Above, 2 Below Street.

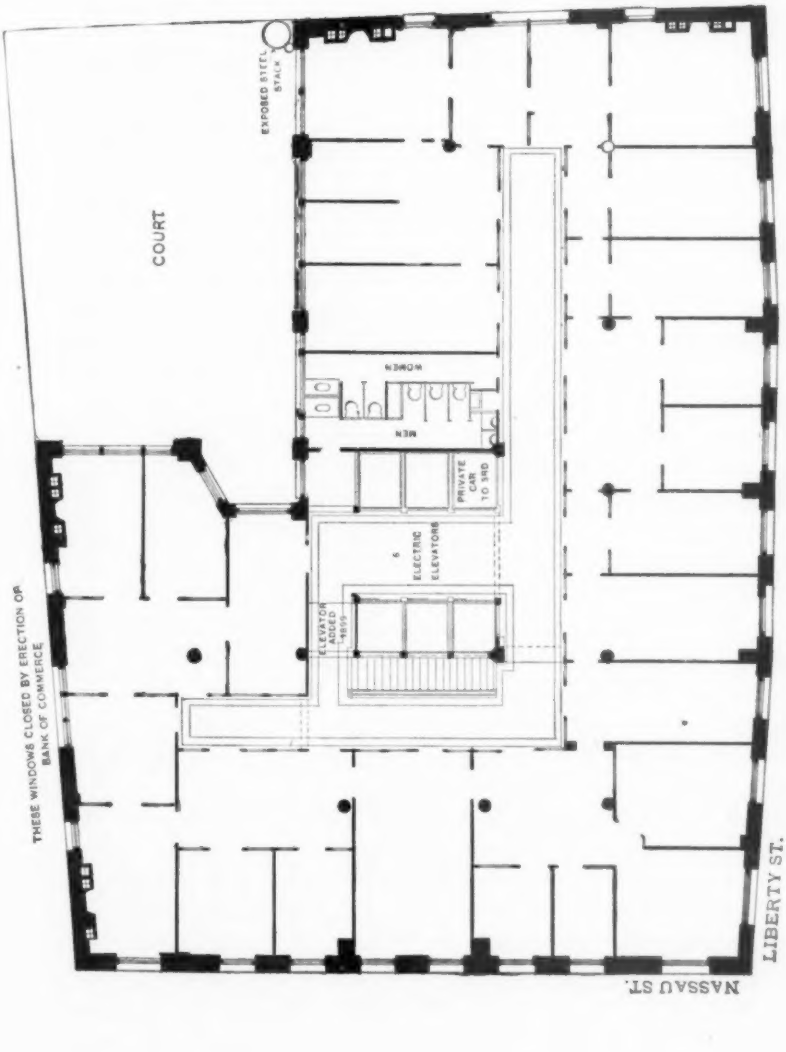


Fig. 489.—German American Building.
15 Stories Above and one Below Street.

after compressing the air, descends to another tank on the fourth level, whence it supplies the flushes of the fixtures below, and is thus used three times over, reducing the water bill of the building by $1,100 per annum.

37. The sanitary systems of New York tall office buildings are installed under the very excellent regulations of the Department of Buildings, and are under the inspection of their officials. The rules, however, have not altogether advanced with the progress of this class of building and require some amendment to meet the conditions of such extended structures.

38. The provisions as to proportions of vent pipes are based on the number of floors, ignoring the number of fixtures, so that one vent line may be serving, say, 14 fixtures vertically superimposed, while its neighbor of same proportions may have to serve the same number plus a complete basement toilet with 20 or 30 fixtures. Extremely tall vertical stacks of soil lines are permitted without regard to what will occur should a stoppage take place. They should be divided into sets not exceeding 80 feet drop. The use of the Kenney flushometer has widely extended since its first adoption in the Bowling Green Offices (16). The Department requires a sub-division of water pressure upon these appliances, which is easily arranged by separate tanks at intervals of 4 to 8 floors.

39. W. C. fixtures are now universally of the syphon discharge pattern. Urinals are no longer permitted to have automatic flushes, although the regulation has not been made retroactive, and many older buildings still continue to waste city water by these appliances. All piping is carried out in screwed galvanized work with extra heavy galvanized fittings.

40. The proportion and location of toilets is a matter of wide variation in practice. Some architects have favored the central toilet-room situated in a middle floor and placed in charge of a regular attendant. This, as in the case of the Lords Court, Fig. 492, and Bowling Green Offices, Fig. 488, adds considerably to the local elevator travel, and the same attendance applied to independent toilet rooms on each floor, can maintain them in equally good condition. The practice seems to have settled down to the latter arrangement.

41. As to the number of fixtures provided, the variation is equally wide, and evidently more a matter of guesswork or imitation than of any attempt at methodical proportion. This will

probably continue so to be, so long as most architects consider the sanitary work of such buildings as a matter with which they are technically capable of dealing. The majority of such installations in these buildings have been designed by plumbing contractors for the architects.

42. The installation of restaurants in these buildings is common, and the introduction of their cooking appliances adds to the steam consumption quite considerably, and also presents problems of the prevention of the smell of cooking, etc., extending into the building. In the German-American Building (15), Broadway Chambers (18), and Dun Building (15), Figs. 482 and 493, the kitchen is vented by a fan drawing from over the range and discharging up a vent shaft, in the latter cases around a chimney stack. In the Bowling Green the vent is discharged into the boiler flue, as well as the gases of the ranges.

43. The lighting of these buildings is universally electrical, and is, as regards distribution and generation, a simple operation. Much of the interior of some buildings requires artificial lighting, see the shaded parts of Fig. 484. The risks taken by many of the owners of such buildings, of having much of their natural light cut off by the erection of equally lofty neighbors, is a matter which has only seldom been taken into account in proportioning the plant.

44. Such cases have been, however, far from uncommon. In the German-American, Fig. 489, 60 windows, and in the Hudson Building, Fig. 490, no less than 120 windows have been closed by neighboring buildings, and this loss of light has had to be partly made up by an increase of artificial light. The lower outside offices of such buildings as line narrow streets like Exchange Place, such as the Broad Exchange (20), Wall Exchange (25), and part of other buildings, require artificial light at all times.

45. Direct-current systems are employed, the feeders being made three-wire, only in order to provide for eventual possible connection of the public service; from the distribution centers on each floor the circuits are carried in two-wire work to the outlets. The regulations of the New York Board of Fire Underwriters and of the Department of Buildings are followed, and are carefully drawn and are now in practical harmony.

46. Since the Bowling Green installation, in which the author used unlined, plain enameled iron conduit, and for the purpose of evading the then regulations, installed therein lead-covered con-

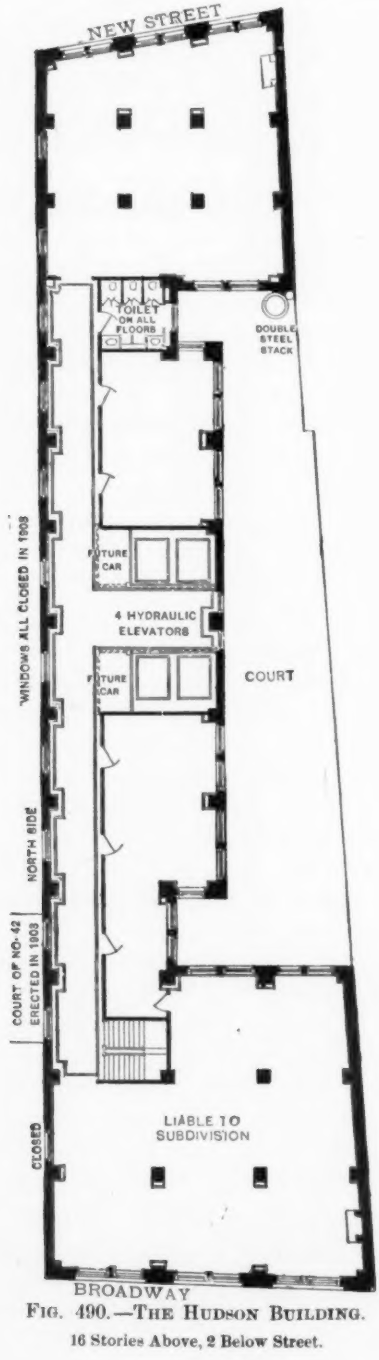


FIG. 490.—THE HUDSON BUILDING.

16 Stories Above, 2 Below Street.

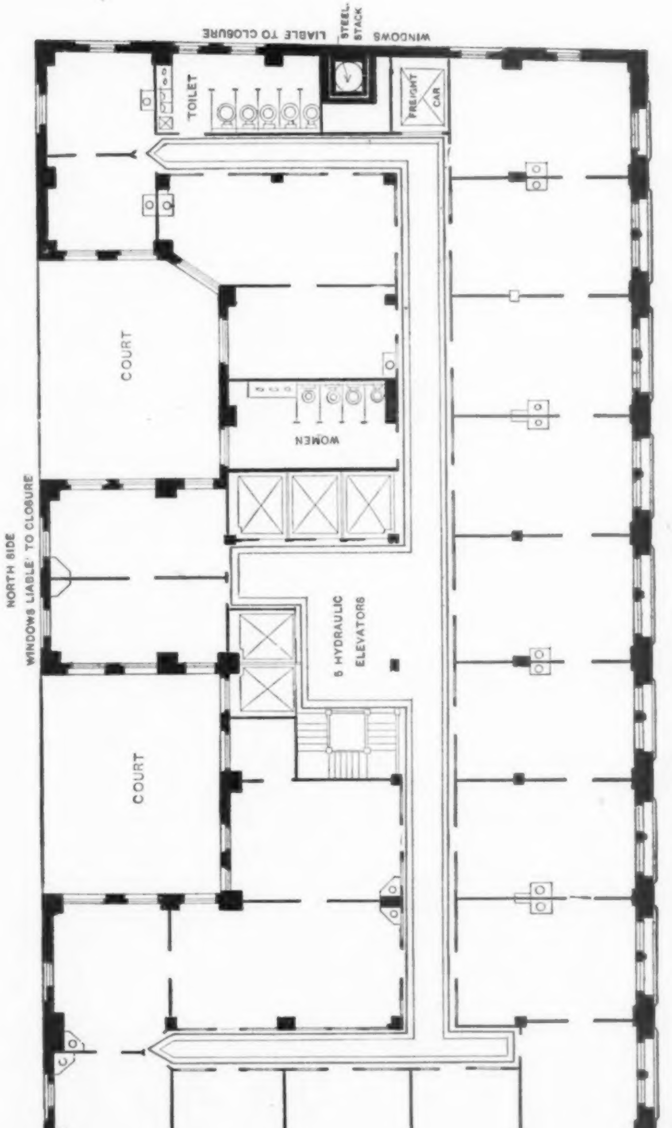


FIG. 491.—CENTRAL BANK BUILDING.

15 Stories and Club Premises Above, 2 Stories Below Street.

ductors, the rules have permitted the use of such conduit and more recently of the flexible steel unlined conduit, which has been widely utilized. The latter offers much advantage to contractors in ease of bending to suit all conditions, but is liable to misuse by workmen and is so rapidly erected that due inspection becomes difficult.

47. Several buildings have the author's vertical system installed, in which the distribution centers are in the cellar and the circuits are extended vertically up the lines of superimposed outlets. About 30 per cent. less conduit and conductors is used in this method, but the fixtures in any room or set of rooms must be cut in or cut out individually. Where that feature is not required, however, the system presents advantages in interior drainage and in the accessibility to the engineer of the fuses of every circuit.

48. The usual practice is to adopt a pressure of 125-120 volts, but in the Dun Building (15), and in a few others, a pressure of 240-220 volts is maintained, operating elevator and other motors in parallel with lighting work. The reduction in size of conductors is not material, but there is a gain in simplicity of service and of connections.

49. The generators are in duplicate or more, generally from 75 to 150 kilowatts capacity, operated in parallel. Consideration of head room sometimes induce the adoption of small units, and the same factor has prevented the adoption of the vertical engine as a motive machine. The compound engine has not made much headway, more for similar reasons of space, than of disregard of its economy. The 4-valve engine is a type well adapted to the conditions, and has found some favor recently. The direct connection of engine and generator is the common form, and has much advantage in absence of noise and vibration. The subject of foundations for these engines, interfered with as they often are by the nature of the buildings' footings, is one in which the designing engineer finds much cause for anxiety.

50. Reference may be made, as instances of difficulties overcome, to the Hudson Building, where three 75-kilowatt machines are mounted parallel on a common concrete foundation 16 inches thick by 9 feet wide, also to the German-American Building, where four engines, 100 kilowatts, are disposed around a column, each foundation being located in a pocket between the grillage beams, and all tied together over the cross-grill by I-beams extending from one to another of the concrete blocks.

51. The description of all the foregoing services leads to the source of power, viz., the chimney, the boilers and the fuel.

By reason of their height these buildings are provided with unusually proportioned stacks. Such proportions as 42 inches to 48 inches diameter, to 240 and even 280 feet high are not uncommon, and the ordinary rules of proportion and work do not apply with accuracy. The discharge of gases is greatly accelerated by the height and by the protection of the stack from heat loss, by its being contained in a warm flue. This practice is general, although in some cases exposed stacks are provided, as in the Bowling Green, 225 feet and 5 feet diameter, which has carried over 700 horse-power besides some ventilation, a kitchen range, and a 60 horse-power boiler of a neighboring building. The desirability of close bricking in of such chimnies is questionable, but is common. In the Broadway Chambers, Fig. 487, 258 feet by 44 inches, the flue is so proportioned that the stack is accessible by means of a boatswain's ladder, and is also built with angle iron joints, so that it could be withdrawn in sections to the roof. The locating of these chimnies is often done with poor judgment, and with resultant loss to their efficiency.

52. With such heights, however, strong drafts are generally available, and the burning of small anthracite is common and effective. The regular practice is to use a No. 3 quality which, with a draft exceeding seven-tenths at the boiler damper, can be carried from 5 to 8 inches thick with five to eight hours' cleanings.

53. Shaking grates are commonly employed, chiefly that known as the McClave. One or two attempts have been made to utilize soft coal in the Hawley down-draft furnace. While excellent results may be obtained, the fact that New York is not a soft-coal center, and that hard small coal is always more readily available, militates against the extension of the use of bituminous coal.

54. Storage of fuel is always limited, the only building where as much as 600 tons can be taken into store being the Bowling Green Offices, 720 horse-power. Some buildings are very inadequately provided, and are hard pressed in case of coal shortage, strikes or even severe weather. In some cases managers have paid $9, and in one case $20 a ton for a supply in such emergencies.

55. The usual supply is by contract for daily or weekly delivery at prices varying from $2.30 to $2.80 per long ton. Automatic stokers are installed in only a few buildings. The strong

draft renders practicable the use of fuels as cheap as above stated, with hand firing, and the size of the plants does not render possible any serious reduction in the labor employed in the fireroom.

56. The water-tube boiler, of one or other of the well-known

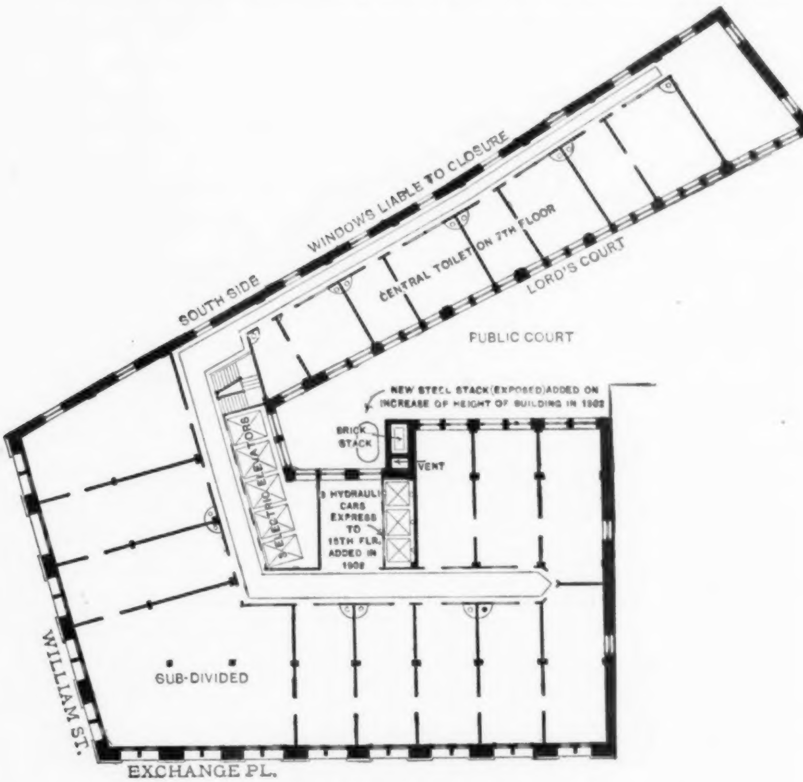


FIG. 492.—LORDS COURT.

15 Stories Above and one Below Street. 4 More Stories Added in 1902.

forms, is in general use in these buildings. Their selection and even their proportions have as often been a matter of available space as of the duties to be performed. Installation of unequal sized units has been found economical in the Central Bank, 480 horse-power, and Hudson Building, 500 horse-power. In the Broadway Chambers, the space made available by the architect's plans best suited the vertical type, and two such boilers are installed, of 200 horse-power each.

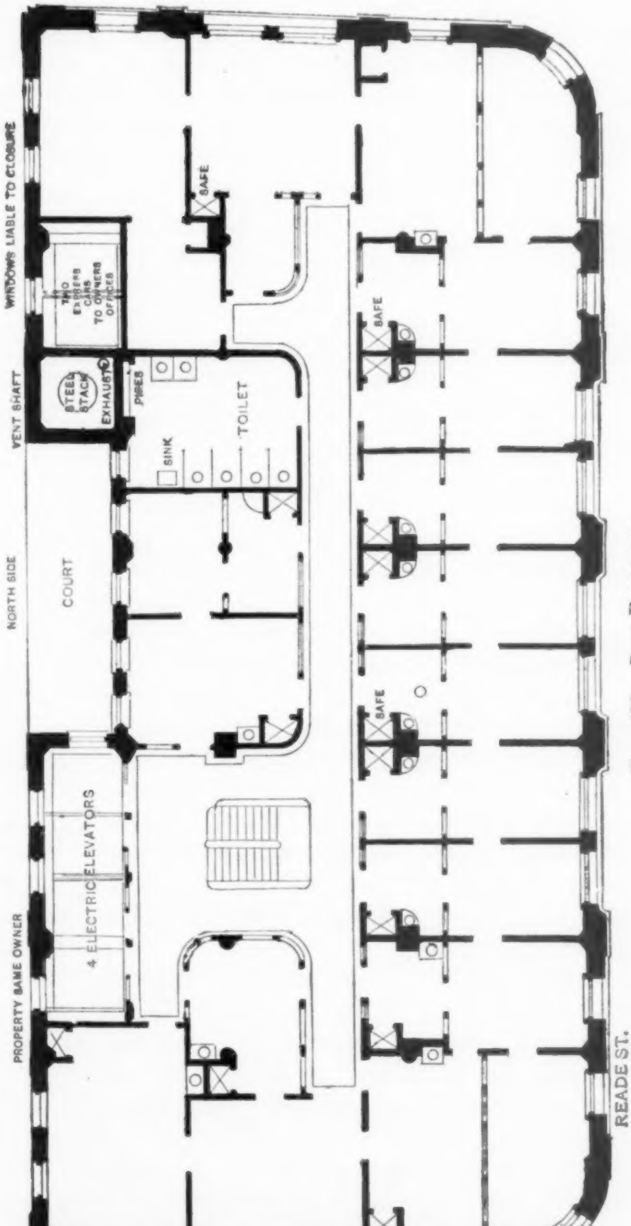


FIG. 493.—DUN BUILDING.
15 Stories Above, 2 Below Street.

57. The matter of piping is one of great importance, but is, remarkably, one in which no regulation by public authority is made, nor, it is regrettable to say, is due regard paid to it by architects and owners. Consequently, it is rare to find power piping and connections with which the operating engineers do not have trouble, and to which changes have had not infrequently to be made. Such occurrences will continue so long as contractors are engaged by architects to do the work of planning this important part of a great building's construction and operation.

58. It must be added, with regret, that the architectural profession as a body have failed to realize their responsibility in regard to this matter of the employment of proper engineering skill upon the work of the design and proportion of the power plant of these large buildings. The annoyances to which they have subjected the owners of such properties by this course of action, annoyances which are peculiarly inherent to systems of mechanical apparatus, and particularly of piping, have led to the present condition of the design and erection of tall office buildings, which has largely been taken out of the hands of the architects and absorbed by general contracting firms. They employ a staff of subordinates, architectural as well as engineering, whose ability is stimulated by haste and competition, but whose methods naturally avoid the introduction of improvements which may add to the prime cost, which alone concerns their employers.

59. To this unfortunate condition the tall office building power plant has been brought, and the present state of the art of their design may be said to have stagnated during several years, their most commendable features being imitations of others.

60. The description of these plants would be incomplete without some reference to the personnel to whom is committed their operation and maintenance.

The operating engineers are licensed by the special Bureau of the Police Department, as are the firemen on all boilers exceeding 10 pounds working pressure. The examinations are not very remarkable for uniformity or scientific accuracy, but suffice in the hands of old and experienced mechanics to discriminate between incapacity and ability.

61. Many of the men holding first-class licenses are old seagoing engineers, the work closely corresponding to their former duties. They are, taken together, an exceedingly intelligent and well-informed class, with very capable and definite opinions on

the apparatus they are set to handle, and the finest sense of scorn for those who have either condemned them to handle ineffective appliances, to work in unwholesome spaces, or to be afforded inefficient assistance and unsuitable fuel.

62. The staff generally consists of a chief, whose pay varies from $100 to $125 per month, one or more assistants at $85 to $100 per month, with firemen and oilers at $60 per month.

A tribute must be paid to these humbler members of the great brotherhood of engineers, whose labors maintain the constant services of these large buildings, on which so considerable and important a population depend for business operations in comfort and security, and who thus maintain them, often under circumstances of great physical discomfort, and even of injury, and amid difficulties which only those who have witnessed their capabilities in sudden emergencies can fully appreciate.

DISCUSSION.*

Mr. William H. Bryan.—The author's severe criticism of the electric elevator may be to some extent justified if limited to tall office building service, which may be said to be the last stronghold of the hydraulic elevator. It is doubtful, however, whether the practice west of New York will confirm the author's extreme views. It is true that we have no extremely tall office buildings in St. Louis, but there are several instances of 12 to 16-story office buildings, in which drum electric machines running 350 to 400 feet per minute are giving satisfactory service. The more recent structures are all equipped with electric machines and in one case where an addition was made, two electric elevators are running satisfactorily alongside of the four original hydraulics. Even if it is admitted that at the present moment the hydraulic system is preferable in tall office buildings, where speeds exceeding 400 feet per minute are necessary, the fact should nevertheless be emphasized that for practically every other kind of service the modern electric machine is displacing all other types.

The reasons are not far to seek. First, Greater economy of power or fuel, which saving in practice is something like one-third, often more. Second, Greater simplicity of installation, one set of generating units serving for light, power and elevator ser-

* For further discussion on this paper, see the discussion appended to paper No. 1036.

vice. Third, Greatly reduced first cost of generating units. Fourth, Greatly reduced space occupied.

While a storage battery improves the electric service when elevators and lights are operated together, it is by no means necessary, and would scarcely be recommended solely for this purpose. If ample engine and generator capacity are provided, with close regulation, large feeders, and low starting current, good elevator service can be secured without seriously affecting the lights.

The cost of running (paragraph 18) seems magnified. In St. Louis we have flat rates as low as $10 per month, and I know of no case exceeding $50 per month. Power rates for such service average 4 cents per horse-power hour.

It may be conceded also that at the present moment the record favors the hydraulic in the matters of safety, reliability and low repairs, although this is strenuously denied by many engineers and manufacturers who are in a position to know the facts regarding the actual performances of the best machines of both types. The electric elevator is still comparatively new, and has passed through a severe experimental process during the last ten years. Its most satisfactory form, and the one which may now be said to be approaching a standard, has only been developed a few years. It is reasonable to suppose that the difficulties and troubles which accompanied the earlier machines have to a large extent been remedied, and that the record for the next ten years will compare well with that of the hydraulic system, if, indeed, it does not equal or surpass same in the still doubtful matters of safety, reliability, and low cost of repairs.

In St. Louis electric elevator machines located overhead have been very successful. This greatly economizes space and cables. Care must be taken to secure pent houses of ample area, well-lighted, and easily accessible. In this respect we are now securing the coöperation of the best architects.

*Mr. R. P. Bolton.**—The discussion seems to have been mainly concentrated upon my condemnation of the electrically operated elevator in paragraph 17, and preceding, which is not, however, practically disputed by any of the speakers in the discussion.

Mr. Byran cites instances of satisfactory operation of electric drum machines at speeds up to 400 feet per minute in 16-story buildings. I do not doubt that, but if the speed of 600 feet were

* Author's closure, under the Rules.

attempted as required by New York conditions in buildings of 16 stories and over, the results would be as I have stated.

Mr. Bryan recites the familiar favorable assertions of general economy, simplicity and reduced cost and space of electrical elevator apparatus, making no distinction between the schedule plant and little installations of one to three machines.

The old contention that one generating unit will serve with economy, for both light and elevator power, is not to be borne out if satisfactory service of both is required.

If such a unit is to be employed for both services, then the size of cylinder must be predicated upon the maximum lighting it is to carry, to which must be added the elevator load at its highest or starting point, and when the latter irregular item is not present, then the cylinder is too large for its work, and is, in the majority of cases, even with automatic cut-off engines, working wastefully. The additional cost of this increase of engine capacity should be debited to the electrical elevator. The cost of outside service in St. Louis, of 4 cents per horse-power hour, is much lower than is obtainable in New York, but even with that low rate, the figures of $10 to $50 per month per elevator, serves to show that the service is less than one-half of that expected of the machines, the cost of which, I quoted in paragraph 18.

To Mr. Nistle's remarks, I reply, that the inherent fault of all electrically operated machines, for rapid schedule service, is lack of direct means of arresting the momentum of the moving elements of the apparatus, except by such mechanically unreliable devices as brakes, whereas the hydraulic cylinder has a direct and positive control on the heaviest parts of the moving apparatus, and this is a feature that the drum electric machine does not possess, and which the Sprague screw machine set out to imitate.

Mr. Rockwood's reference to the "Plunger" type of elevator, introduces it as one which he apparently regards as suited to all sorts of conditions, inclusive of office-building work. I cannot be accused of any hostility to this exceedingly desirable form of apparatus, inasmuch as I was the only engineer who had the courage to adopt it for a height of travel exceeding 200 feet, and to undertake the responsibility of boring in Manhattan rock to the depth required.

But I did not introduce it in my paper for the reason that it is not a suitable apparatus for high office buildings, as it lacks the same element as the electric, namely means of positively arrest-

ing the momentum of travelling parts. It is true that this is the case only on the up-run, but that is the side on which the plunger system is weak.

With reference to Mr. Suplee's remarks, the only hope for ventilation in tall office buildings, is in the employment of ventilating engineers in advance of or on equal terms with, the architect. By the time any engineer gets to work on these buildings the conditions have been fixed so far as to render any effective system of ventilation impracticable.

I am completely in accord with Mr. Gifford's remarks in favor of the four-valve engine for this class of work. I have not favored, nor have I installed any compounds on office building work. In hotels, the conditions are very different and warrant their use.

I would have been glad to adopt Mr. Colles' suggestion, of a table of cost of maintenance and charges per floor, above 16, and hope I may be able to do so at a later date. It will exhibit, I believe, the blundering system by which very many of the buildings have been equipped. I quite agree with Mr. Colles that the electrical part of the equipment of office buildings is a matter of very simple character. Outside of the vertical circuit system, paragraph 47, and of the abandonment of interior lining of conduits, paragraph 46, no improvement of any kind has been made in ten years past in this direction.

No. 1046.*

THE STEAM TURBINE IN MODERN ENGINEERING.†

BY W. L. R. EMMET, SCHENECTADY, N. Y.
(Member of the Society.)

1. Most of you are more or less familiar with the general character of the steam turbines of Parsons and DeLaval, which are the principal forerunners of the type to which this paper relates. These machines represent the opposite limits of possibility in steam turbine design, and a comparison of them is, therefore, very interesting and suggestive in considering other methods.

2. The expansive force in steam is capable of imparting to the steam itself very high velocities, and the problem which presents itself to the designer of steam turbines is to devise means by which the velocities of moving jets or columns of steam may be made to give up their work effectively to mechanical parts which run at practical speeds.

3. In the DeLaval turbine the total available power of the steam is used in a single set of nozzles, and all the useful work obtained is delivered by jets from these nozzles to a single circle of moving buckets. In the Parsons turbine there are 30 or 40 circles of moving buckets and stationary directing vanes. A small proportion of the energy in steam is used in each set of stationary vanes to impart velocity to the steam with the most effective direction. This energy of motion is then in part given up to the next circle of moving buckets. These turbines appeared at about the same time, nearly twenty years ago. One does its work in one process and the other in many successive processes, but

* Presented at the Chicago meeting (May and June, 1904) of the American Society of Mechanical Engineers, and forming part of Volume XXV. of the *Transactions*.

† For further discussion on this topic consult *Transactions* as follows :
No. 345, vol. x., p. 680 : "Notes on Steam Turbine." J. B. Webb.
No. 648, vol. xvii., p. 281 : "Steam Turbine." W. F. M. Goss.
No. 876, vol. xxii., p. 170 : "Steam Turbines." R. H. Thurston.
No. 987, vol. xxiv., p. 999 : "Steam Turbines from Operating Standpoint." F. A. Waldron.

in spite of this wide difference of method, both turbines at an early date produced good results, and results which were not widely different from each other in degree of thermal economy.

4. It might be supposed that the existence of these results, accomplished in such widely different manners, would have quickly led to the development of other practical methods and to the earlier introduction of the steam turbine on a large scale, but in fact many years elapsed before either of these turbines began to be very extensively developed, and before any important new types made their appearance. The Curtis steam turbine, which has very recently appeared on the market, is the first radically new commercial machine different in type from either the Parsons or DeLaval machines. The general purpose of the Curtis design is to produce results with a reasonable number of simple parts and at moderate speeds, while the Parsons turbine requires a very large number of small parts and the DeLaval turbine employs excessively high speeds inapplicable to mechanical purposes without the use of speed reducing gearing.

5. Mr. Curtis's experiment on steam turbines had been going on for about four years before the General Electric Co. undertook to build any machines for service. At the end of that period, and at a time now about four years ago, the author first became connected with the enterprise, being asked to express an opinion concerning the value of the invention. Other engineers had reported unfavorably, and the discontinuance of work was a possibility. The opinion given was that the invention afforded great possibilities, particularly in the matter of simplicity and economy of production; that the development of commercial machines was justified by the experiments and should be begun at once, and that the development of high degrees of steam economy was to be expected with further experience.

6. This report led to the beginning of work on a larger scale; the first step being to build a 600 kilowatt unit, which was put into operation at Schenectady in November, 1901. This machine was built on the general lines previously considered by Mr. Curtis, with a horizontal shaft and two stages with groups of wheels in separate casings. The arrangement of buckets and nozzles were about as shown in Fig. 494. Tests of the machine, which were carried on for some months showed very good results, which were published in the author's paper to the American Philosophical Society in April, 1902. The success of this machine led to the

undertaking of commercial work on a large scale, and the experience so far obtained and a careful study of mechanical possibilities led to the adoption for this new work of radically new mechanical designs, applied to bucket and nozzle arrangements similar

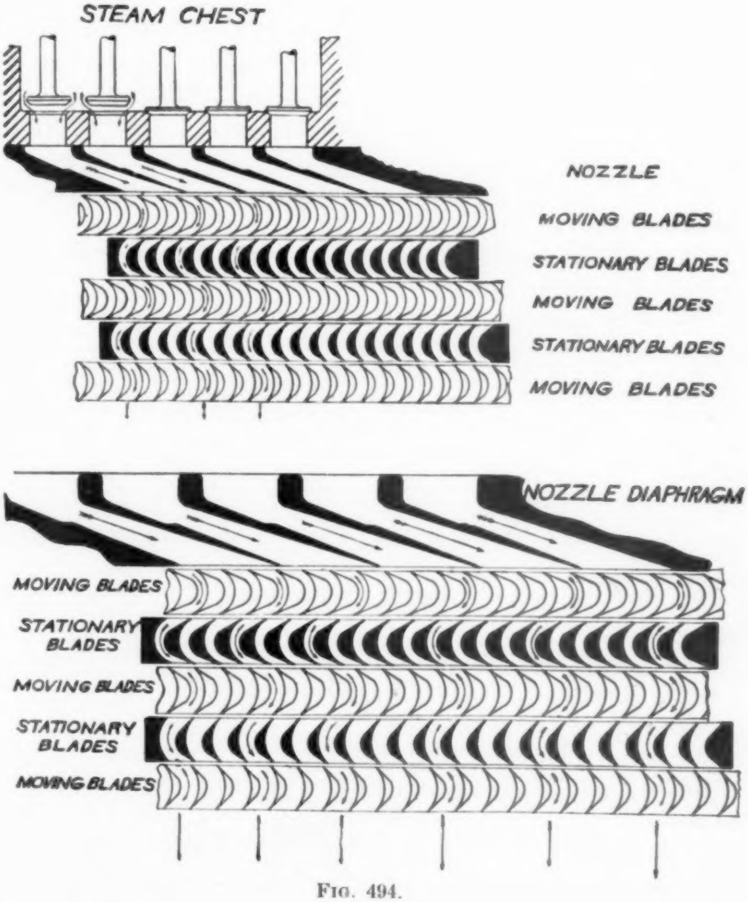


Fig. 494.

to those previously used and first recommended by Mr. Curtis. The first machines designed of this new type were a 5,000 kilowatt unit for the Chicago Edison Co. and a 500 kilowatt unit, the first of which was installed at Newport, R. I. Four machines from the former and about forty from the latter design have up to the present time been installed.

7. These machines are built with shafts in a vertical position; the total weight of revolving part being borne by a step-bearing at the foot of shaft, and shaft being steadied and aligned by three bearings, one at the top of generator, another near the foot of shaft, and a third between the generator and turbine. Many reasons led to the adoption of this arrangement which involved some untried features, but which afforded very great practical advantages. A careful consideration of designs indicated that the untried features involved little risk of serious difficulty, and experience has shown that this judgment was sound. Some of the important advantages of the vertical shaft type are as follows:

The relative positions of revolving and stationary parts are definitely fixed by the step-bearing.

The stationary part is symmetrical, easily machined and free from distortions by heat.

The shaft bearings are relieved from all strain, and friction is practically eliminated.

The shaft is free from deflection and can be made of any size without reference to bearings, which can be placed where convenient and operated with surface speeds, which would not be practicable with the horizontal arrangement.

These features make possible the use of a very short shaft, and consequently the longitudinal spacing of moving and stationary parts is very little effected by temperature differences.

The turbine structure affords support and foundation for the generator.

The cost of foundations is very small, and the solidity and alignment of foundation is not of vital importance.

Much floor space is saved.

All parts of the machine are conveniently accessible.

Failure of lubrication cannot injure the shaft or other expensive parts.

8. Another new feature of these designs is the arrangement of valves; the turbine being governed by the successive opening of steam operated valves, which are independent of each other, but which are all controlled by the centrifugal governor. With this arrangement the speed control of the turbine is not dependent upon the successful operation of all the valves, since the governor automatically keeps open as many valves as the machine requires, and in the event of trouble with one valve opens another to take its place. The character of these first vertical shaft designs is

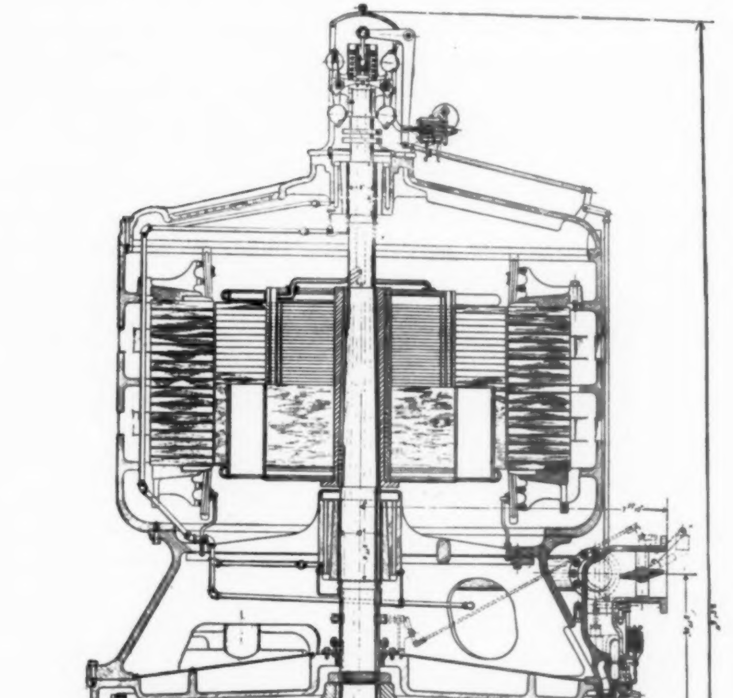


FIG. 495.

illustrated by the cross section drawing of Chicago machine, Fig. 495. Most of the drawings of parts which are here given show the designs which have been adopted in connection with later machines, and do not apply to the machines at Chicago or others of that date. The principles involved are, however, the same in all.

9. In this Chicago machine there are two stages, each stage having a single wheel which carries four rows of moving buckets. On the stationary part there are three rows of stationary buckets opposite each group of nozzles used. The shaft is in one piece from the step-bearing to the top of the generator. The wheels are of cast-steel mounted upon it with taper fits. The stages are separated by a cast-iron diaphragm which is fitted with valves which are now operated by hand, but which are being arranged to operate automatically; it being advantageous to constantly maintain a certain pressure relation between the stages.

10. The step-bearing is shown by Fig. 496 and consists of two cast-iron blocks, one carried by the end of the shaft and the other held firmly in a horizontal position and so arranged that it can be adjusted up and down by a powerful screw. The lower block is recessed to about half its diameter, and into this recess oil is forced with sufficient pressure to balance the weight of the whole revolving element. The amount of oil required is small. About 5 gallons per minute is used in the 5,000 kilowatt machine, but with a good alignment it could be satisfactorily operated with a much less amount. The oil after passing between the blocks of step-bearing wells upward and lubricates a step-bearing supported by the same casting. This whole structure is inside of the base, and a packing is used between the oil chamber and the base, so that oil or air cannot get into the vacuum chamber. A small steam pressure is maintained between the sections of this packing, in order that these objects may be accomplished with certainty. In many cases these same step-bearings have been operated with water instead of oil, in which case no packing is necessary; the water being allowed to pass into the base. In some of our latest designs water will be used exclusively; the lower surface of step-bearing being of wood and no packing being provided.

11. The extreme conditions to which these step-bearings are subjected, and a complete lack of precedent for such designs led at first to many doubts concerning the success of this feature. Experience has, however, shown that these doubts were without

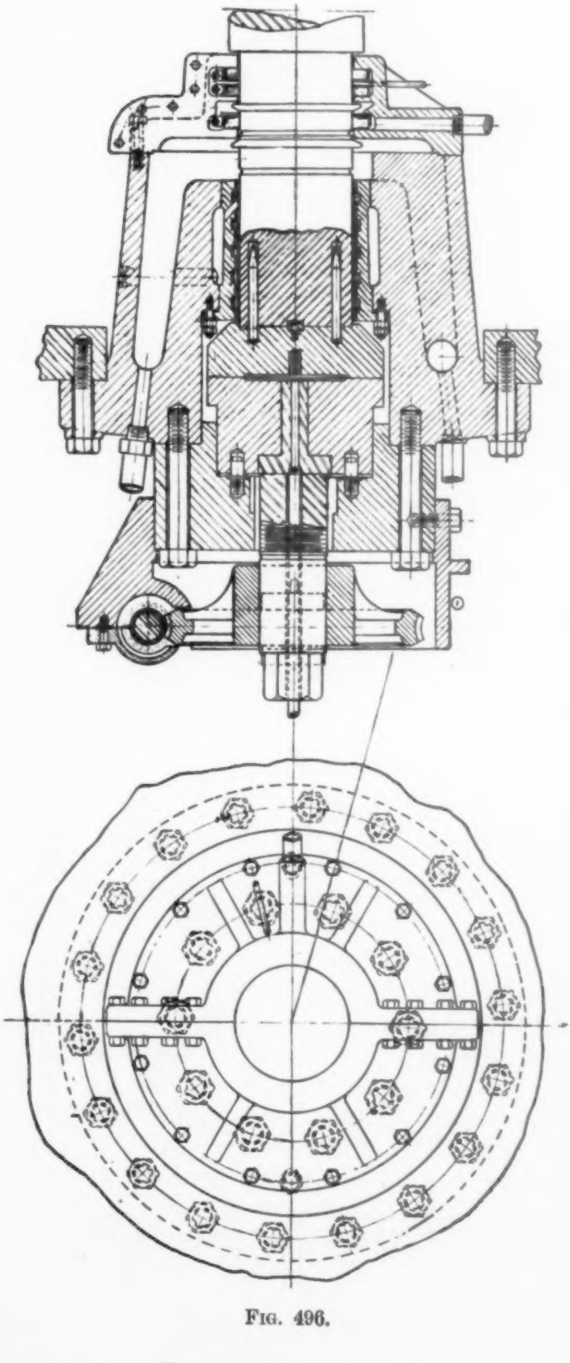


Fig. 496.

foundation. Practically no troubles or interruptions have resulted from this cause, and the step-bearings have shown a ruggedness and stability far beyond our expectations. Many of our turbines have been operated under more or less temporary conditions in incompleted stations, and there have been many accidental stoppages of step-bearing pressure while machines were running. In hardly any cases have such stoppages caused even an interruption of service. The step-bearing surface cuts immediately when lubrication is stopped, but the metal from it is removed very slowly and it has the power of re-establishing itself almost immediately when oil flow is again started. We expected that this condition would exist in smaller machines, but did not hope for it in larger machines, and took precautions to provide accumulators and other auxiliaries necessary for permanent maintenance of step-bearing pressure. In spite of these fears and precautions the step-bearing pressure has through accidents failed five or six times in the Chicago Edison plant, and all of the three machines now operating there are running on step-bearings which have been subjected to such stoppages. In no case has any harm resulted to the machines, and the bearings have always operated just as well as if they were in a perfectly new condition. Our policy in adopting the vertical design and putting our dependence in the step-bearing was to put all our eggs in one basket and watch the basket. We have now discovered that very little watching is necessary. In our newer designs we are providing a powerful brake bearing on the lower surface of a chilled iron ring carried by the lower wheel. This brake can be conveniently operated from the outside, and can be used to take the whole weight of the revolving part in case the step-bearing support should fail. In ordinary operation the shoes of this brake will be set about 0.01 inch below the brake ring. It is thus in a position to receive the revolving part in case the step-bearing support should fail. Another and more important function of this brake is to stop the machine when it is desired to do so. One of the 5,000 kilowatt vertical shaft machines will run for four or five hours after the steam has been shut off, unless load is put upon it or a brake is applied.

12. One of the most important matters in all steam turbine work is the matter of balance, and the importance of good balance applies as well to vertical turbines as those that are operated in a horizontal position. When the balance is good the bearings on vertical turbine shaft are practically free from strain or fric-

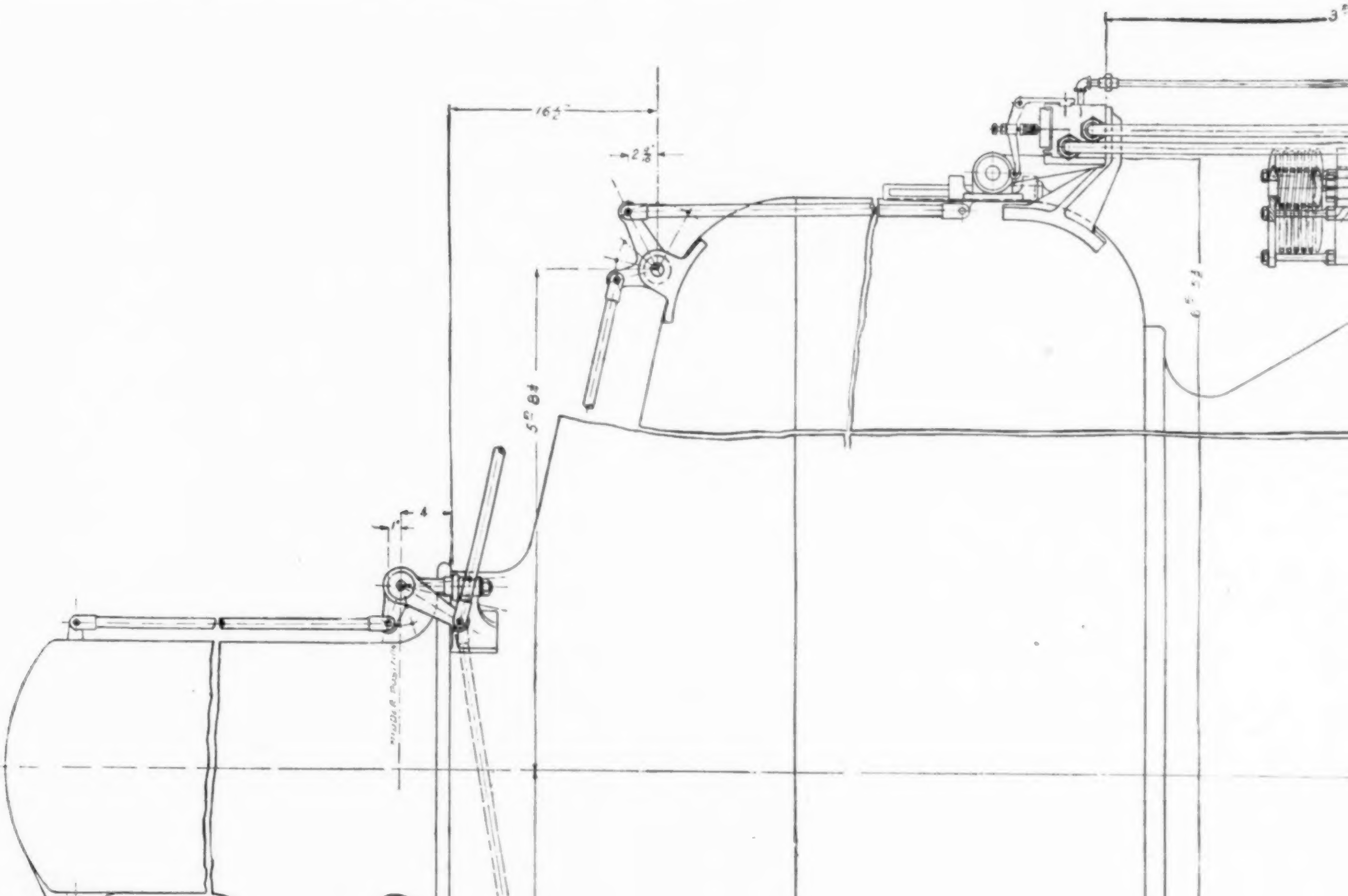
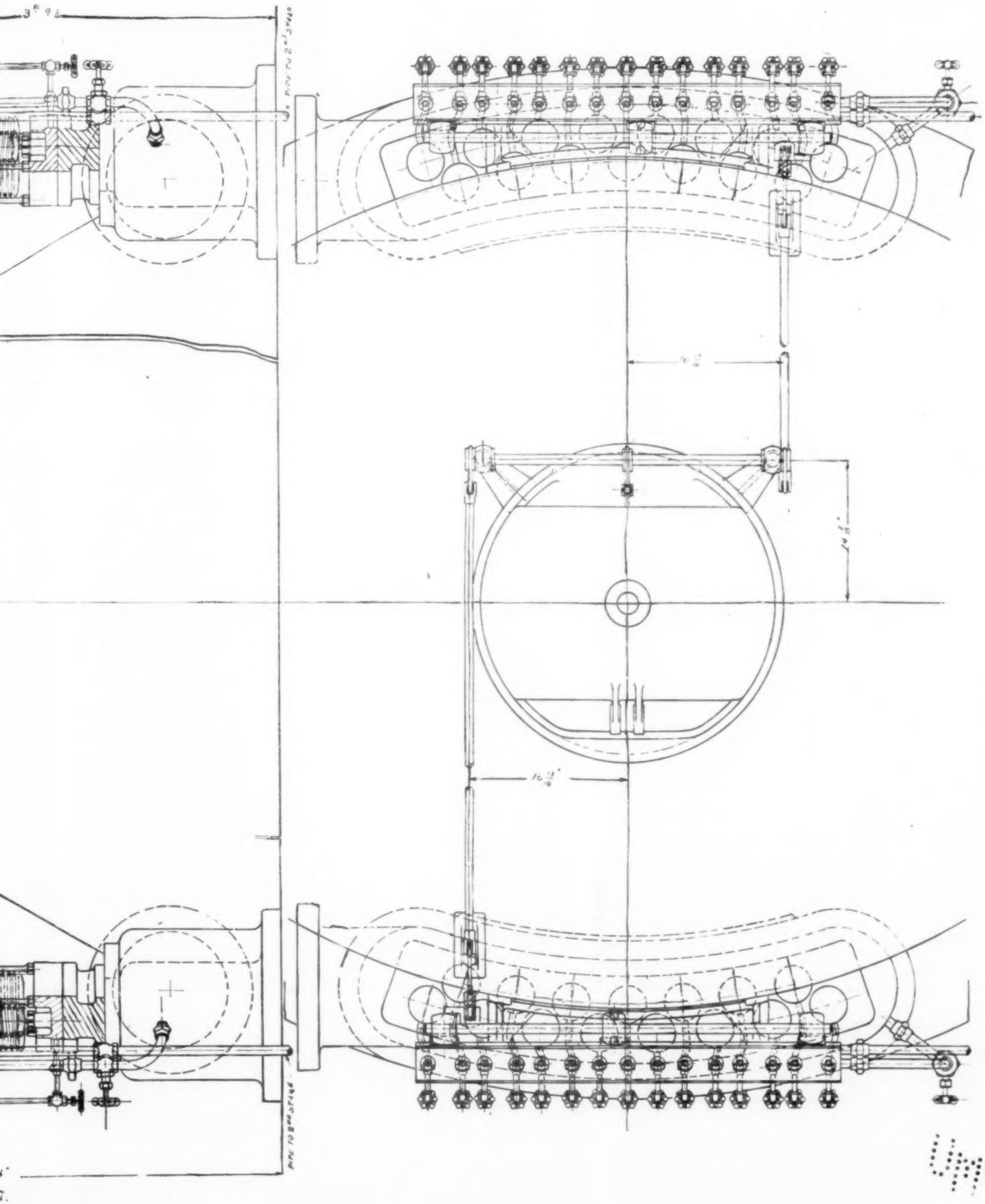


Fig. 497.



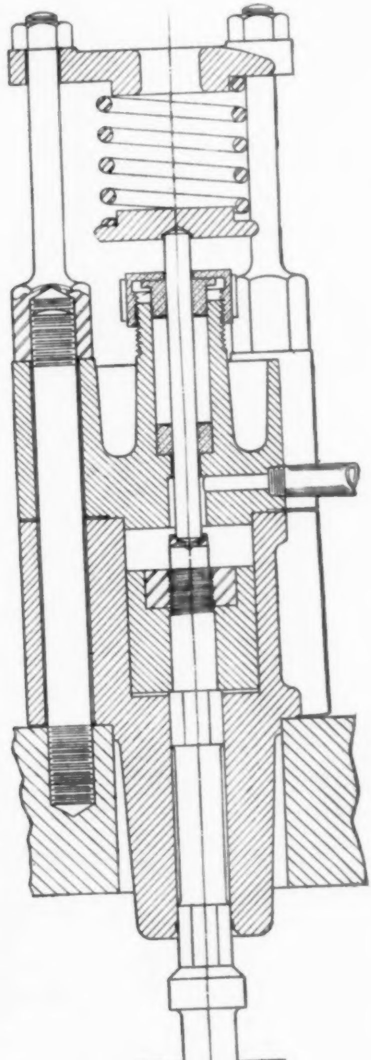


FIG. 498.

tion. It is possible to operate these machines successfully with very considerable imperfection balance, but a perfect balance is practicable, and should be attained in every case.

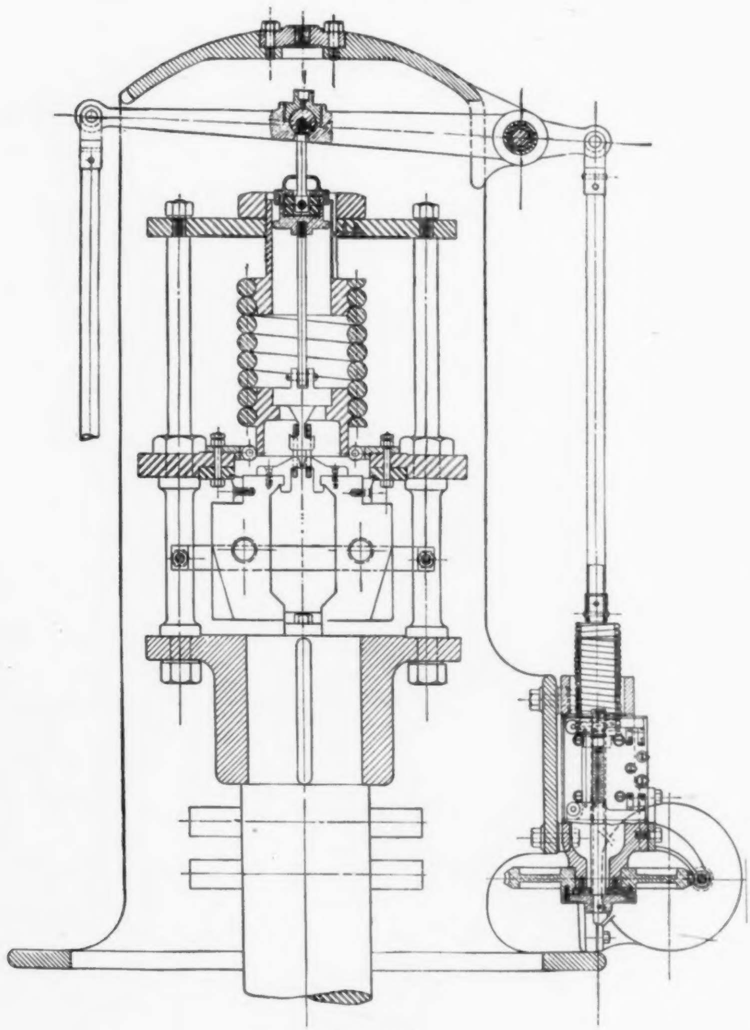


FIG. 499.

13. Fig. 497 shows the connection of valve mechanism to governor in one of our new 5,000 kilowatt turbines. Fig. 498 shows one of the controlling valves used with this arrangement. Each of

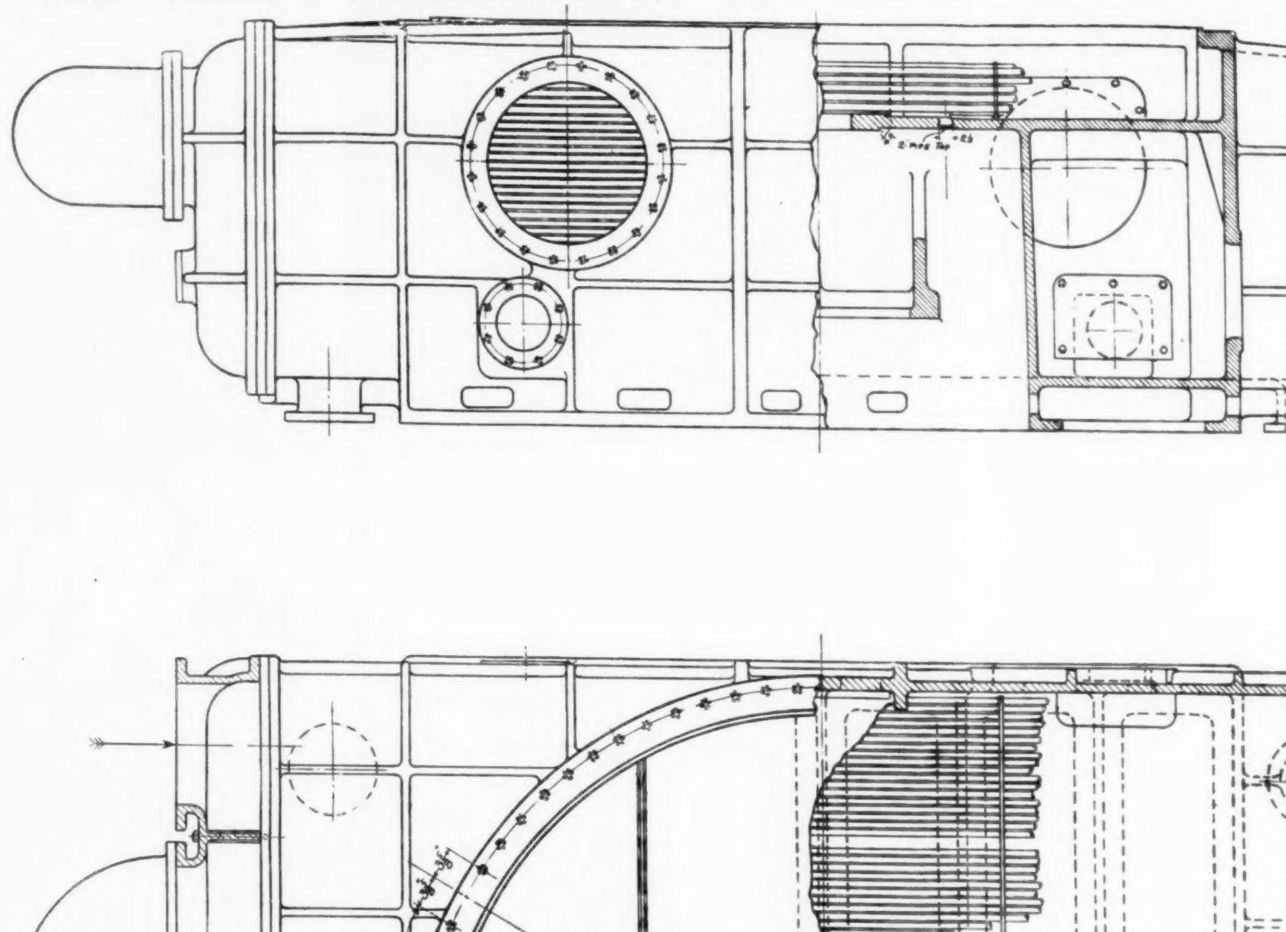
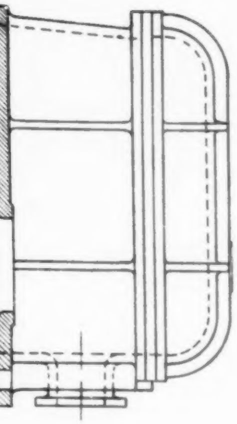
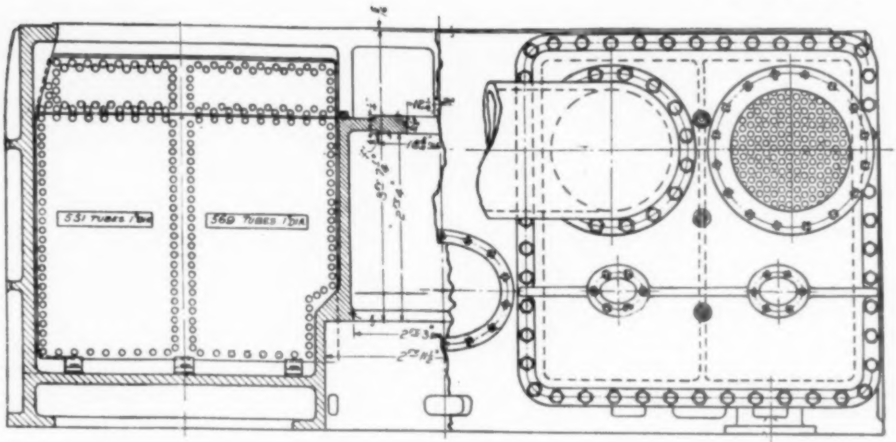
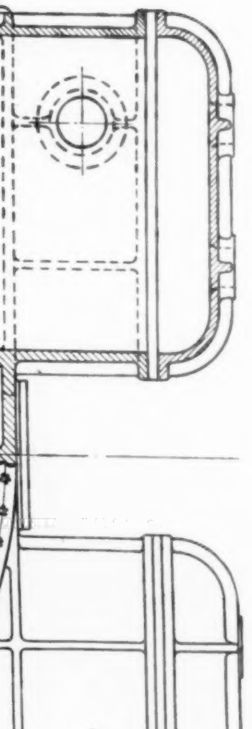


FIG. 500.







500.

these controlling valves operates one or more main valves which communicate with the first stage nozzles. This controlling valve is so designed that it always passes positively from one of its seats to the other. No matter how gradually the force is applied it opens both ways on the principle of a pop safety valve. It is thus always firmly seated and is free from the deterioration which leakage would cause. The total number of these valves required imposes a light load upon the governor which is made strong enough to give any desired accuracy of speed regulation without the possibility of lag or sticking. The controlling valves on our earlier vertical shaft turbines were actuated by electric magnets. This mechanical arrangement is simpler and more positive.

14. Fig. 500 shows a base for supporting one of our 5,000 kilowatt turbines which also performs the functions of a surface condenser. This combination has been adopted in several of our newer designs; among them the 2,000 kilowatt unit, which is now installed at the St. Louis Fair. This arrangement is adapted to the production of the best possible vacuum with a given water supply, and also has the advantage of a considerable saving of space. Two 5,000 kilowatt units equipped with these condenser bases are now being installed for the Boston Edison Co.

15. Fig. 501 shows the cross section assembly of the 500 kilowatt unit above mentioned. This machine is designed with two stages and three rows of buckets per stage. Its general features and characteristics are similar to those of the larger units, and need not be separately described. This machine has been tested under a variety of conditions at Newport, and has given results which illustrate very well the advantages of the type. Among other tests that were made the machine was operated on a rapidly changing railway load; the momentary variations of load amounting to about 120 kilowatts. In one test the average load carried with this fluctuation was 250 kilowatts, and the steam consumption was 24.4 pounds per kilowatt hour output with saturated steam. Another test was run with similar fluctuations and with an average of 421 kilowatts. The steam consumption under this condition was 20.7 pounds of saturated steam per kilowatt hour output. The best reciprocating engine under conditions of the first test would probably consume from 28 to 30 pounds per kilowatt hour.

16. The 5,000 kilowatt and 500 kilowatt two-stage machines above mentioned are, as I have said, designed in accordance with

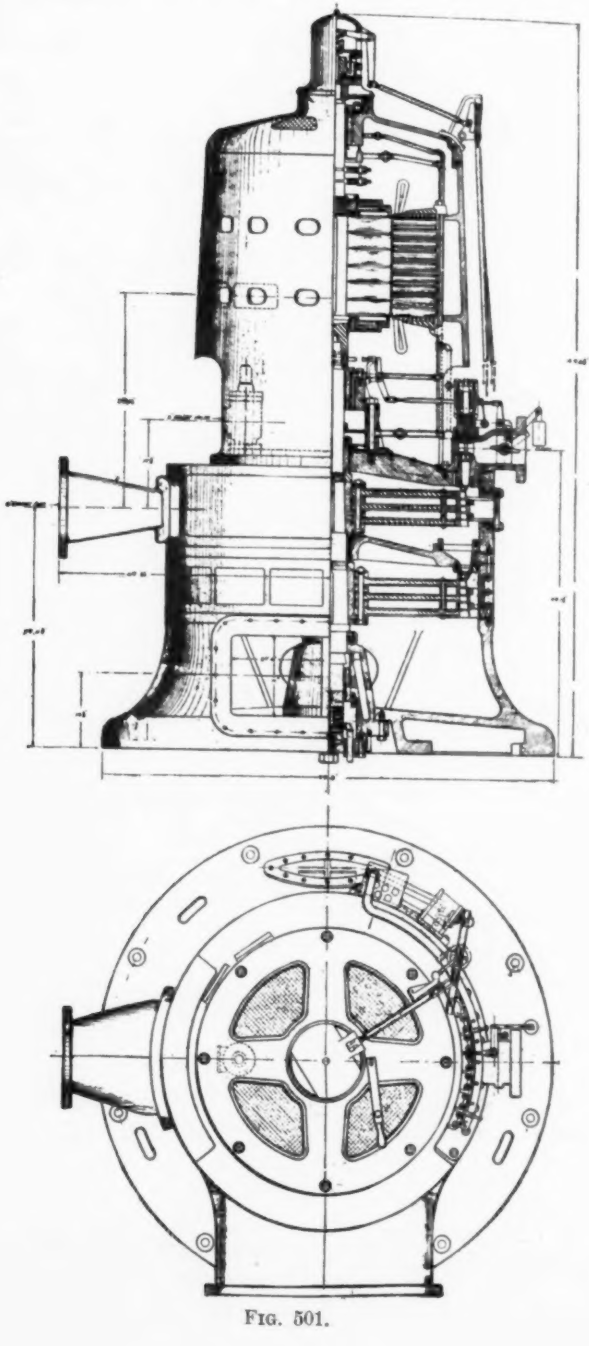


FIG. 501.

Mr. Curtis's original conceptions before we had the benefit of much study and practical experience. After construction of these machines had been begun, we became convinced that other ar-

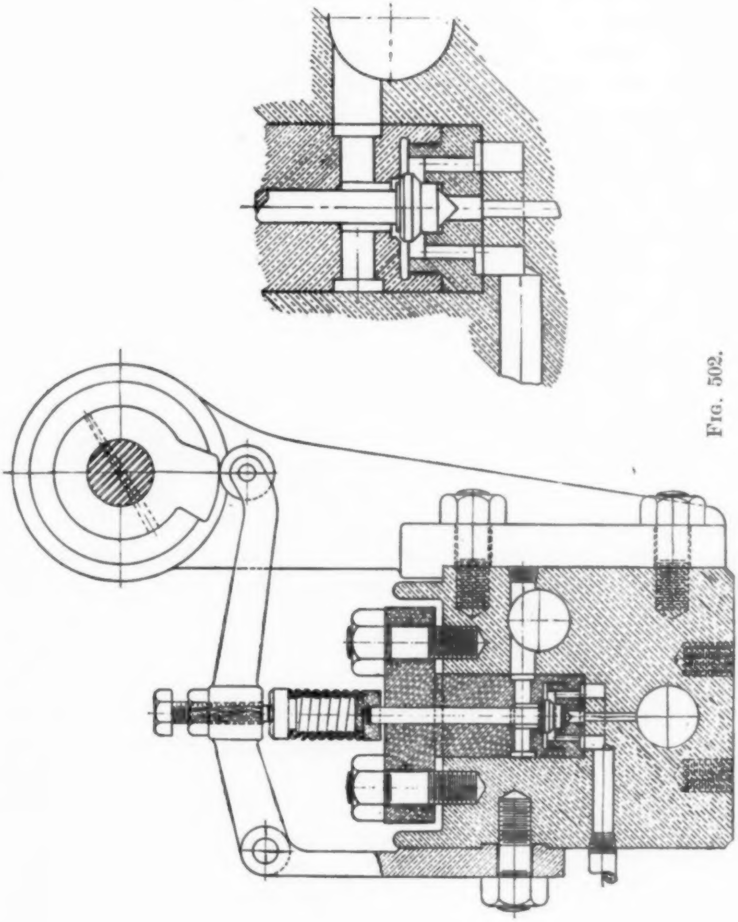


Fig. 502.

rangements of steam parts would be more advantageous, and began the design of a new type, the construction of which was well advanced before our first experiences with the first type were obtained. Theory and experiment indicated that we could obtain better economy from the new design adopted, and it also affords the advantages of greater simplicity and a better mechanical structure. These new machines are ideally simple, with large

clearances and practically no tendency to distortion. No parts of them are subjected to heavy strains, and no particular accuracy or thoroughness of workmanship is required in any part. Machines of this type, which have very recently been put into operation, have started without any trouble or interruption and given very perfect service.

17. After we had embarked upon the manufacture of a great number of large turbines, and had already put into production this new type, we began to receive reports of rapid advances in the turbine art elsewhere. Large turbines of the Parsons type in Europe produced high steam economies, and we began to meet in competition good guarantees of steam economy with relatively low prices. The results of tests on our first machines were, therefore, awaited with great interest. We had good reasons for our beliefs concerning the results which could be produced, but we were very slightly supported by practical experience, and were dealing with a problem which involved immense values.

18. There have been many delays in completing the first of these new machines, and it has been necessary to interrupt tests in order to meet customers' requirements, and mistakes and accidents in steam plant have caused further delay in testing. We have recently obtained accurate tests from this first new machine as originally designed. The capacity of this machine is 2,000 kilowatts. It operates a 6,600 v. 25 cycle generator at a speed of 750 revolutions per minute. It is temporarily installed in the General Electric Company's power station at Schenectady, with a surface condenser having 6,000 square feet of cooling surface. The following are results obtained under different running conditions. The tests on March 12th and on May 11th were made upon different machines of similar design. Considering the different conditions, the results are consistent.

	March 12th.			May 11th
Load in Kilowatts	637	1000	2000	2270
R. P. M.	750	750	750	750
Gauge Pressure	150	160	155	100
Superheat F.	215	242	242	250
Corrected Vacuum	28.2	28.9	28.73	28.1
Lbs. Steam used per Kw. hour	20.1	16.3	15.3	16.2

Such analysis of results as we have been able to make indicates that a different proportioning of certain parts will give us a substantial improvement.

DISCUSSION.*

Mr. Francis Hodgkinson.—Mr. Emmet's paper is of particular interest, insomuch that he describes many features of the mechanical construction of the turbine in which he is interested. It is unfortunate, however, that he has not furnished us with more records of performance made under a variety of operating conditions, instead of but one test on one machine, under what might be described as elaborated conditions of operation, if not under what might be styled a somewhat commercial condition—that is, 28.73 inches vacuum and 242 degrees Fahr. superheat. The results, however, sound very excellent, but the high vacuum and superheat should be reckoned with.

In my paper I describe a test made by Mr. F. W. Dean under somewhat high operating conditions, although not as high as those described by Mr. Emmet. With 180 degrees superheat and 28 inches vacuum, the steam consumption per brake horse-power hour was 11.17. With the superheat 100 degrees, it was 12.06 pounds at almost the same load. If we assume the increase in economy due to superheat, between 180 degrees and 240 degrees to be at the same rate as between 100 degrees and 180 degrees, the result of 11.17 pounds per brake horse-power hour would be improved about 5.7 per cent. The increase of vacuum from 28.08 inches to 28.73 inches will improve the economy of a Westinghouse-Parsons turbine about $2\frac{1}{4}$ per cent. Allowing a generator efficiency of 95 per cent., brings the result of this 400 kilowatt turbine to 14.5 pounds per kilowatt hour, against Mr. Emmet's best result of 15.3 pounds per kilowatt hour with a 2,000 kilowatt unit.

---

* For further discussion on this paper see the discussion appended to paper No. 1037.

No. 1047.*

THE DE LAVAL STEAM TURBINE.†

BY E. S. LEA, TRENTON, N. J.
(Member of the Society)
AND
E. MEDEN (Non-Member).

1. The fundamental principles of the De Laval Steam Turbine are clearly shown in Fig. 503. It is a pure impact turbine, with a single turbine wheel, carrying one row of buckets, to which the steam is delivered in free jets at the highest possible velocity. These steam jets come from stationary nozzles, tapered so as to increase their cross sectional area toward the outlet end of the nozzle, and so calculated that the steam, before leaving the nozzle, has fully expanded down to the pressure prevailing in the exhaust chamber of the turbine, and has assumed a correspondingly high velocity, so that its whole available energy has been transformed into kinetic energy.

2. The velocity of the steam jets varies considerably, owing to change in pressure of the steam before entering the nozzles, to varying exhaust pressure, and to a greater or less degree of moisture or superheat in the steam. The lower limit of this velocity found in general practice might be considered as about 2,000 feet per second, which is obtained at a steam pressure of about 45 pounds per square inch, at an exhaust pressure equal to the atmospheric pressure, and with steam containing 10 per cent. of moisture. The upper limit is found to be about 4,400 feet per

* Presented at the Chicago meeting (May and June, 1904) of the American Society of Mechanical Engineers, and forming part of Volume XXV. of the *Transactions*.

† For further discussion of this topic, consult *Transactions* as follows:
No. 345, vol. x., p. 680: "Notes on Steam Turbine." J. B. Webb.
No. 648, vol. xvii., p. 81: "Steam Turbine." W. F. M. Goss.
No. 876, vol. xxii., p. 170: "Steam Turbines." R. H. Thurston.
No. 987, vol. xxiv., p. 999: "Steam Turbines from Operating Standpoint." F. A. Waldron.

second, at a steam pressure of 200 pounds per square inch, at 27.5 inches vacuum, with the steam superheated 200 degrees Fahr. The velocity of the steam will determine the conditions under which a maximum of the transformation of the steam jets' kinetic energy into useful mechanical work can be reached, these conditions being the same as for impact water turbines.

3. The nozzle angle, or the angle of the steam nozzle with

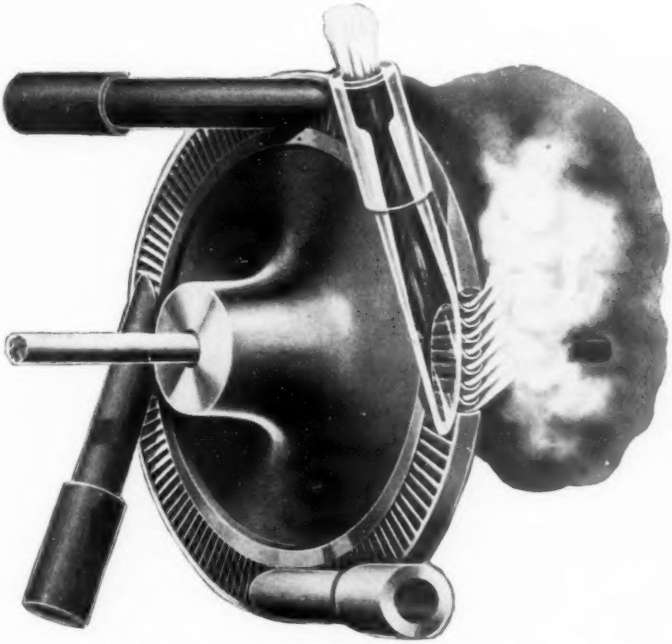


FIG. 503.—DE LAVAL WHEEL AND NOZZLE.

relation to the plane of the wheel, should be as small as possible. A certain mathematical relation should exist between the nozzle angle, the velocity of the steam jet, the peripheral velocity of the turbine wheel, and the inlet angle of the buckets. The outlet angle of the buckets should be the smallest possible. Practical considerations limit to a certain degree the attainment of proper angles for the very best efficiency. Thus, in the De Laval Turbine, a nozzle angle of 20 degrees has been established for all sizes of the turbine, the inlet and outlet

angles of the buckets are made alike, and are 32 degrees for smaller sizes, 36 degrees for larger sizes. With these angles fixed, and taking into consideration the thickness of the buckets, it will be found that the best theoretical peripheral velocity of the turbine wheel will be about 950 feet per second for a steam jet velocity of 2,000 feet per second, and about 2,100 feet per second for a jet velocity of 4,400 feet per second.

4. Contrary to popular belief, there are no reasons, either theoretical or practical, to prevent the building of a safe turbine wheel, with a peripheral velocity as high as 2,100 feet per second; only economical reasons have put a limit to it. In the turbines that have been built, the actual peripheral velocity varies between about 1,400 feet per second in the larger sizes; and about 500 feet per second in the smaller sizes. In comparison with existing machinery and other types of steam turbines, these velocities are exceedingly high, and have necessitated the solution of some very interesting theoretical problems, such as the calculating of the strains in wheels revolving at high speeds, determination of flexible shafts suitable for carrying these wheels, etc. These theories would occupy too much space to enumerate here, and as they have been published in the technical literature, we have omitted them. We would especially refer to a book on the steam turbine "Die Dampfturbinen und die Aussichten der Wärmekraftmaschinen," by Dr. A. Stodola, of Switzerland, where, apart from some slight inaccuracies, the said theories have been published in an exhaustive and able manner. This book is now being translated into English by Dr. Lewis C. Loewenstein, Instructor in Mechanical Engineering at Lehigh University, and will be on the market in about two months. This book also contains that part of thermodynamics, dealing with the outflow of steam through nozzles, and the determination of expanding steam nozzles, as well as the efficiencies obtainable from the work of the steam turbines.

5. The diameters of the turbine wheels are such, in relation to the given peripheral velocities, that the speeds run from 10,600 revolutions per minute for the largest size, to 30,000 revolutions per minute for the smallest size. These speeds are reduced approximately 10 to 1, by helical gearing, giving driving shaft speeds of 900 to 3,000 revolutions per minute. A single gear wheel is provided in the smaller types (see Fig. 504), and in the larger sizes they are double (see Fig. 505). If the larger types

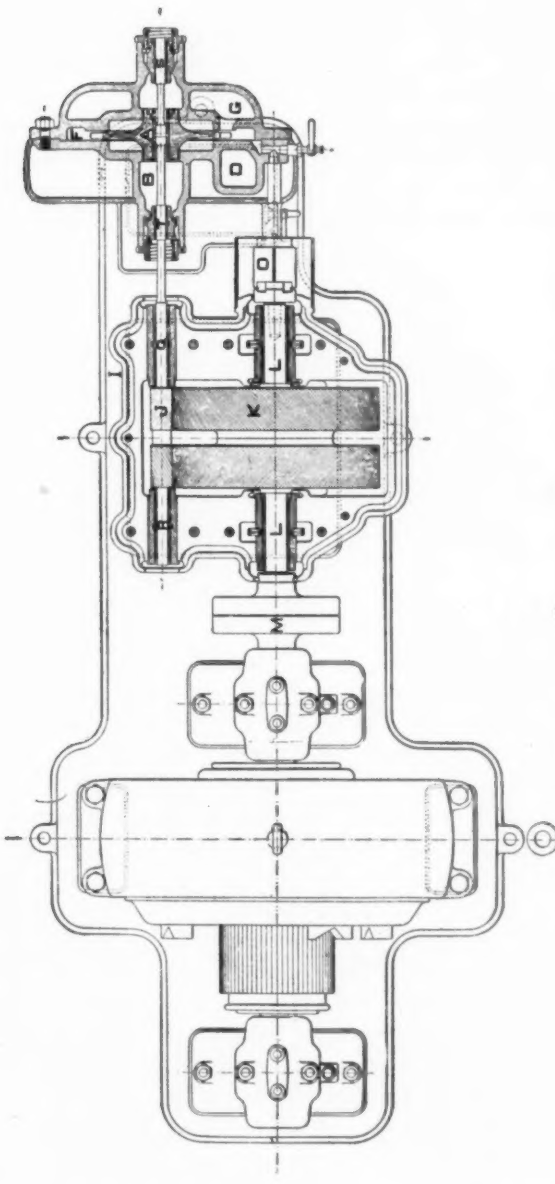


FIG. 504.—SECTIONAL PLAN DE LAVAL TURBINE DYNAMO, 30 H. P.

were single geared, the pressure in the pinion bearings, due to the pressure between the teeth of the gear and the pinion, would be too great at these speeds; therefore, the gears are made double

so that half the load is taken by each wheel, the gear pressure on one side of the pinion balancing the pressure on the other side, thus eliminating the pressure in the pinion bearings.

The characteristic high velocities of the principal parts of the De Laval Turbine also create some interesting practical problems.

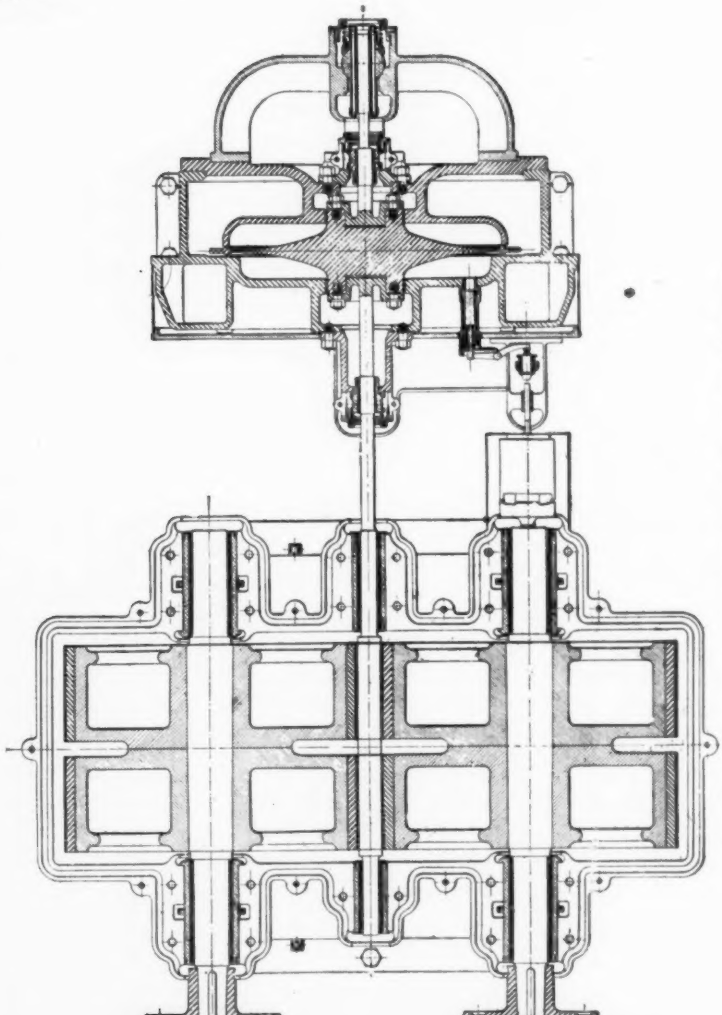


Fig. 505.—Sectional Plan De Laval Turbine, 300 H. P.

6. We will first consider the turbine wheel itself, which is shown in section in Fig. 506. The wheel is designed with a factor of safety at normal speed of about 8, and with radial and tangential stresses due to the centrifugal force constant throughout the wheel. The profile of the wheel is a logarithmic curve asymptotic to the radial axis of symmetry of the wheel section. The buckets, which are inserted into milled slots in the rim of the wheel, when actuated

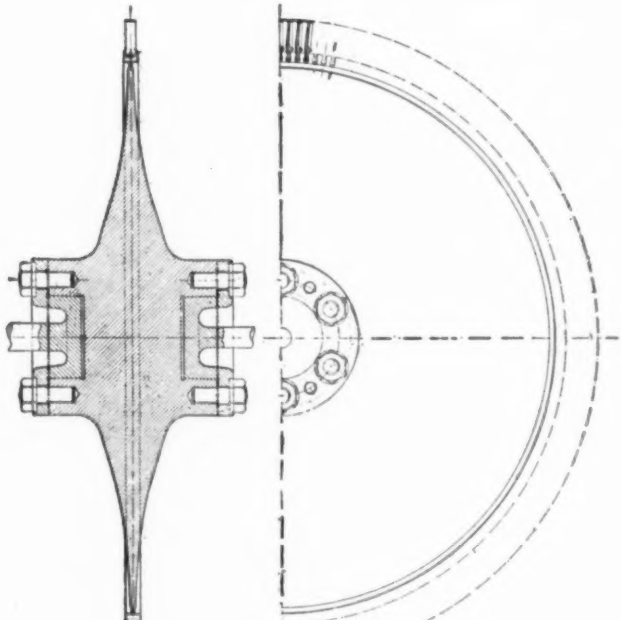


FIG. 506.—SECTION DE LAVAL TURBINE WHEEL.

by centrifugal force, load the solid wheel body at its outer periphery to an amount equal to the centrifugal stresses in the body. The stresses vary with the square of the speed, and with increasing speed they will gradually increase to a point where the wheel will burst.

7. In spite of speed regulating mechanism and safety stops, a motor of any kind might race, as all regulating devices are liable to derangement, and safety stops, which as a rule are seldom used, sometimes fail to operate. This, of course, applies also to

steam turbines. It is therefore necessary to provide means for the prevention of serious damage. In the De Laval Turbine this protection is obtained by reducing the thickness of the wheel close to the periphery, which naturally decreases the strength of the wheel at this point, the stresses here being about 50 per cent. higher than in the rest of the wheel. At normal speed the factor of safety at this point of the wheel is about 5; consequently the wheel will burst here at about double its normal speed, and in such a manner that the rim holding the buckets is broken up into pieces, which, on account of their small size, are unable to do any damage to the wheel case. At the moment the rim leaves the wheel, the stresses in the solid wheel body are considerably reduced, at the same time the wheel becomes unbalanced, and as the clearance between the heavy hub of the wheel and the safety bearings in the surrounding wheel casing is very small, the hub of the wheel will come in contact with the latter, which efficiently act as a brake on the wheel, and bring it to a stop in a short time, as with the buckets gone, the steam has no effect whatever on the wheel. Exhaustive experiments have verified these statements, it having been found that turbine wheels without this decrease in section at the outer periphery, having purposely been speeded up, would burst through the center in two or three heavy pieces, which, at the high velocity, a wheel case of ordinary proportions would not resist. Such pieces have been driven through an experimental wheel case of steel castings, having walls 2 inches thick. With the wheels as made, however, they are perfectly safe, and in the event of the rim being striped, no damage will result except to the wheel itself.

8. As it is possible to design a turbine wheel for any radial and tangential stresses, it might be asked why the wheels are not made so strong that it will be impossible, with the available steam velocities, to run them up to the bursting point. The reply to this is, that it would be too expensive and not practicable to design the rest of the turbine and connected machinery to run safely at a corresponding speed.

9. In this connection we will consider the speed regulation mechanism of the De Laval turbines. This consists of a common centrifugal governor, actuating a throttle valve in the steam supply line of the turbine. With this the pressure can be closely controlled, but not entirely shut off. In most cases, though, sufficient to prevent the turbine going above its normal speed when

running light. This is especially true of turbines running non-condensing. In condensing turbines operating with very high vacuum, the passive resistances are sometimes extremely small, and even if the governor valve throttles the steam considerably below the atmospheric pressure, the remaining pressure may be sufficient, at no load, to increase the speed above the normal. To prevent this speed increase, a second regulating mechanism is provided, the purpose of which is to decrease the vacuum in the wheel case. This apparatus consists of a small valve which is directly actuated by the governor, but only after the governor valve in the steam line has been shut off. This valve either lets air into the wheel case, decreasing the vacuum, or in such cases where the vacuum in the condenser must be maintained for other machines, it admits air into a regulating valve mechanism placed in the exhaust line of the turbine. When air is let into this valve, it more or less shuts off the communication between the wheel case and the condenser, thereby raising the pressure in the wheel case, which then increases the passive resistances of the wheel, and checks the expansion of the steam in the nozzles, and, together with the steam throttle valve, holds the speed within the normal limits. In case of accident to the governor valve mechanism this air valve will also effectually prevent destructive racing.

10. The peripheral velocity of the gear wheels is about 100 feet per second. The pinion is made of high-grade high-carbon crucible or nickel steel. The gear wheels are made of soft steel of low carbon. The teeth are carefully generated at an angle with the shaft center and the pitch is very small, insuring a smooth contact with a minimum amount of noise. The noise cannot be entirely eliminated, but with great care in cutting the teeth, and giving close attention to alignment and center distances, it has been possible to reduce it to a minimum and to a point where it is in most cases of no consequence. The gears are continually lubricated, but with a very small amount of oil. If they get the proper amount of lubrication and care is taken that no sharp grit, such as cement dust, coal dust, or the like, is allowed to enter them, they will operate for many years without visible wear. The gears are encased as much as possible, to prevent the entrance of dust or foreign matter. The gear wheels were originally made of bronze, but it soon developed that this material, as a rule, became crystallized after about two years of continuous operation, when pieces of the teeth were broken off and destroyed the gears.

Steel gears have now been in operation for about nine years, without showing any of the disadvantages of bronze.

11. Little is to be said about the bearings. They are all lined with white metal. The low-speed bearings for the gear shafts are similar to bearings for electrical machinery of same speed, and are provided with ring oilers. Ring oiling, on the other hand, has not proven to be satisfactory for the high speed bearings. The turbine wheel shaft usually vibrates slightly, which is communicated to the oil rings, they then refuse to follow the shaft, and consequently do not furnish proper lubrication. It is also found that the temperature of the oil in this case will increase too much, and drip lubrication has been found more satisfactory, only a small quantity of oil being required. With the high speed it is very important that the lubrication should not be interrupted, as it takes but a short time for the bearing to run hot. Wick lubrication has so far proven the most reliable. It must, however, be arranged so that the oil leaves the wick tube in drops, and with a sight glass below the tube through which the amount of feed can be ascertained. The oil is filtered by the wick, which insures clean oil in the bearing, and the oil will flow as long as any oil remains in the tank. With oil tanks of ample size there will not be much attendance required. It seems, though, in the present advanced stage, that opposition is sometimes met with in having this method of lubrication used. The common sight-feed lubricator with such a small number of drops as are required, has the disadvantage of a very small opening for the oil, so that a small amount of dirt will suddenly interrupt the lubrication. The bearing will then immediately heat. Any mechanical arrangement for forced lubrication is in itself more or less apt to get out of order. It is all right for slow-speed machinery, which, in case of interruption of the oiling, can run a considerable time on the oil already supplied, and until the trouble can be discovered and remedied; but it is more or less uncertain for high-speed apparatus.

12. It might be interesting to touch on the practical difficulties which the De Laval Steam Turbine, like any other radically new machine, was compelled to meet, after it had been put on the market. The turbine naturally had its troubles from defects due to faulty material and workmanship, but these have been remedied. There have been troubles with bearings becoming overheated. This was partly due to faulty workmanship, but in

many cases it can be ascribed to the lubrication, either to failure in keeping the oil reservoir filled, or else to the sight-feed lubricators, which in themselves might have caused trouble. As more machines have been put on the market, they have become more fully understood, and are therefore receiving better attention; consequently these troubles have been gradually reduced. Furthermore, there has been trouble with the buckets. It has sometimes happened that one or more buckets have broken, and come out of the turbine wheel, but without doing any further damage. Generally the turbine, after losing a bucket, can be continued in operation, as the turbine shaft is sufficiently flexible to take care of the unbalancing, though it is best to take out the turbine wheel and replace the buckets. The only explanation of these troubles is that the buckets are subjected to vibratory strains of more or less unknown origin, as their ability to withstand centrifugal force and the action of the steam jet is amply sufficient. In the smaller sizes, below 100 horse-power, broken buckets have been very rare. In the larger sizes, it has been somewhat more frequent. Although the causes of bucket breakage are not yet accurately determined, it has been possible to remedy the trouble where it has occurred. One cause of the undue vibrations of the buckets may have its source in the turbine wheel itself, which, if not homogeneous, will, under action of the centrifugal force, expand unevenly in different directions, thereby unbalancing and causing vibration of the wheel at full speed. This trouble has been overcome by replacing the wheel. The buckets are also subject to more or less wear due to the action of the steam. The cause of this is also very difficult to determine. It may be that the buckets are chemically affected and that thin films of oxide are blown away by the steam, or it may be caused by mechanical wear due to small solid particles coming with the steam, such as rust, or scale from the pipes. It may also be due to some electrical phenomena. However this may be, it is a fact that wear takes place and it is very doubtful that it can be entirely prevented. It has been found in a few cases that buckets have been worn out in a year, necessitating replacement. In other cases the wear has been very slight, even after a run of four to five years. The wear affects only the steam inlet side of the buckets, and will only increase the steam consumption to a slight degree. In tests made on a turbine of 100 horse-power, where the edge of the buckets had been worn away about one-sixteenth inch, the steam

consumption was about 5 per cent. higher than with new buckets. The wheel and buckets are, however, so designed that an insertion of a new set of buckets can be easily made at a small cost.

13. In looking about for proper fields of usefulness, the De Laval Steam Turbine, in common with other steam turbines, first developed the direct connected electrical unit, no difficulty being met in adapting both direct and alternating current generators for direct connection to the gear shafts at their moderate speeds. In many cases De Laval Turbines can also be used, with advantage, for belt transmission.

14. However, the field where the De Laval Turbine is particu-

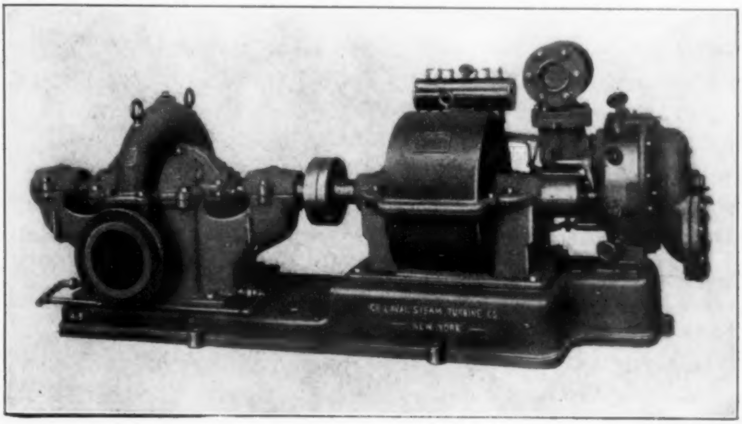


FIG. 507.—55 H. P. TURBINE PUMP.

larly suitable is in connection with centrifugal pumps; these pumps require certain determined velocities to enable them, at a given lift and water quantity, to give the best efficiency. With the De Laval Turbine it is easy to produce the most suitable velocities; with the small turbines, having one gear shaft, for all lifts from 15 feet to 150 feet, and with the large turbines, with two gear shafts, for lifts from 40 feet to 300 feet. These velocities are ofen difficult to obtain with other steam motors. Fig. 507 shows a 55 horse-power single stage turbine pump.

15. For a greater lift the centrifugal pump has been directly connected to the high-speed turbine shaft. The pump wheel will then revolve with a velocity of 10,000 to 30,000 revolutions per minute, depending on the different sizes. The pump wheel will

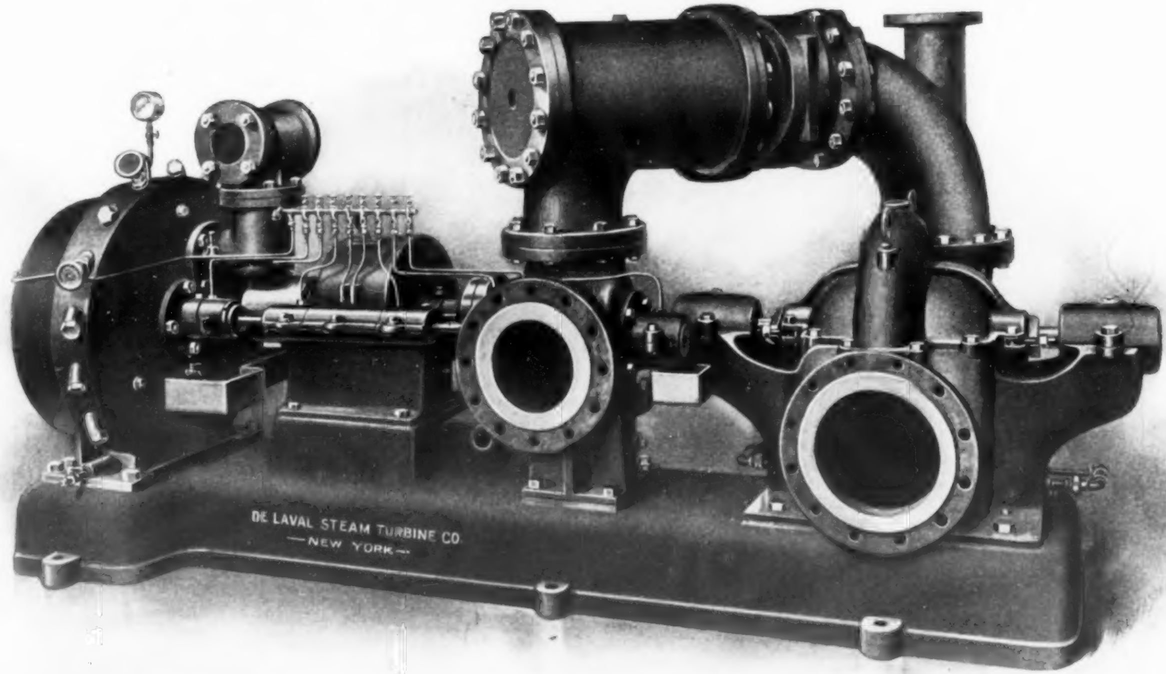


FIG. 508.—HIGH PRESSURE CENTRIFUGAL PUMP.

naturally be very small, and will not produce any suction, but must be fed with another pump, which is connected to the gear shaft, running at a considerably reduced velocity. This latter pump sucks the water and presses it into the high-speed pump wheel, which then gives the high pressure required. Pumps of this type have been made for lifts up to a normal head of 850 feet on a single wheel, which, at a decreased water quantity, can go up to 1,000 feet, the small pump wheel giving an efficiency of about 64 per cent. They have in some cases been made, and are in operation, as boiler feed pumps. A pump of this type is shown in Fig. 508.

16. Another field where the De Laval Turbine is well adapted is for direct connection to blowers for all pressures above 4 inches water, for which a blower can be practically built. The high velocity of the turbine being particularly suitable for this purpose.

17. About the steam consumption; it is difficult to make any general statements. It varies for the same size turbine with the steam conditions, in about the same manner as for other steam motors, but the degree of variation can be considerably different for the various sizes of turbines, dependent upon the diameter and speed of the turbine wheel. It may be sufficient to give here a few diagrams showing the steam consumption of different sized turbines under more or less favorable conditions.

18. Fig. 509 shows the result of a test of a 10 kilowatt non-

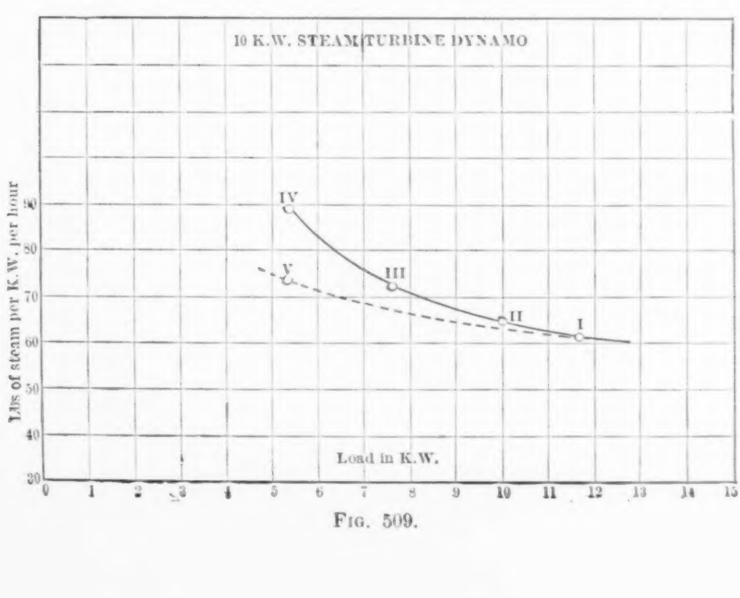


Fig. 509.

condensing turbine dynamo. The boiler steam pressure is 140 pounds per square inch; steam dry saturated. The curve I, II, III, IV gives the steam consumption per kilowatt hour with all steam nozzles open during the varying conditions of the load, the governor valve alone having to take care of this variation by throttling the steam pressure. The curve IV shows the steam consumption with the nozzles shut off in proportion to the varying load. In this manner the nozzles will be supplied with steam of the full steam pressure at all times.

19. Fig. 510 shows a test made on a 30 horse-power steam turbine

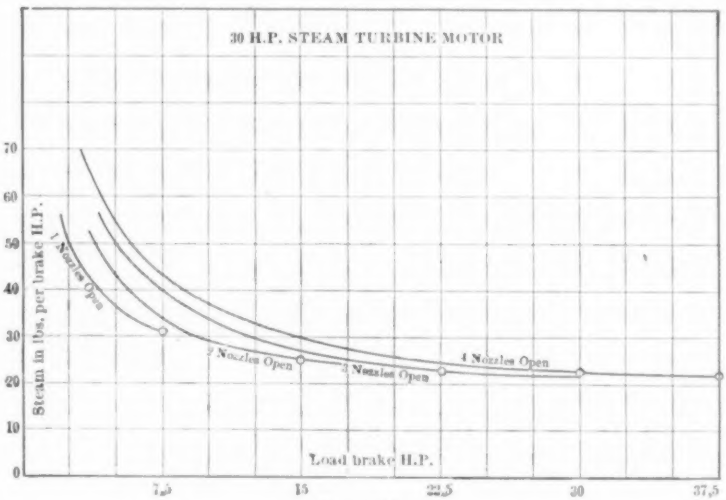


FIG. 510.

motor, condensing, with $25\frac{1}{2}$ inches vacuum, steam pressure being $125\frac{1}{2}$ pounds per square inch above the governor valve. Steam; dry saturated. The four different curves show how the steam consumption per brake horse-power varies with the varying load, with and without regulating the number of nozzles opened.

20. Fig. 511 shows a test made on a 30 horse-power steam turbine motor, non-condensing, with different steam pressures above the governor valve, with the nozzles suitable for the different steam pressure. The curves represent respectively 35, 50, 75 and 100 pounds boiler steam pressure per square inch. The number of nozzles opened have been varied according to the varying load.

21. Fig. 512 shows the result of a test made on a 300 horse-

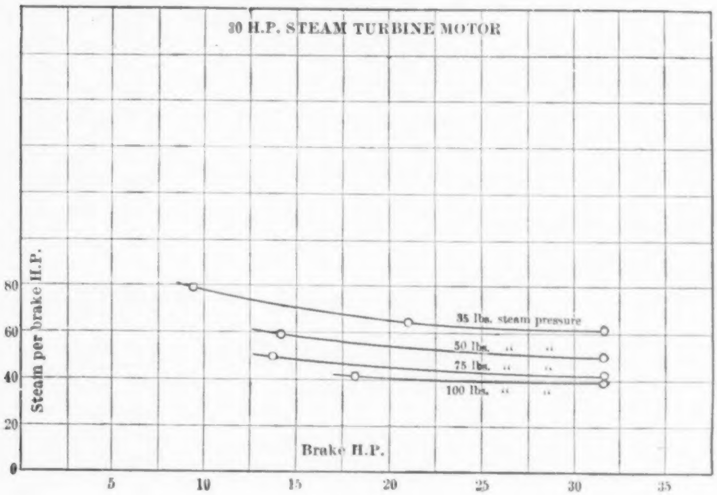


FIG. 511.

power steam turbine, steam pressure about 200 pounds per square inch, vacuum about 27 inches. The curve I, II gives the steam consumption for dry saturated steam. The curve III, IV gives the consumption for superheated steam. The superheat varied from 90 degrees Fahr. at maximum load to about 20 degrees Fahr. at the smaller loads.

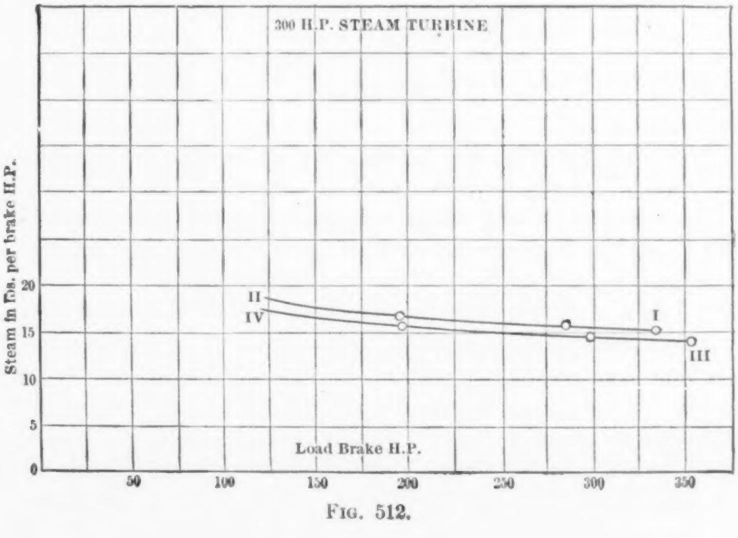


FIG. 512.

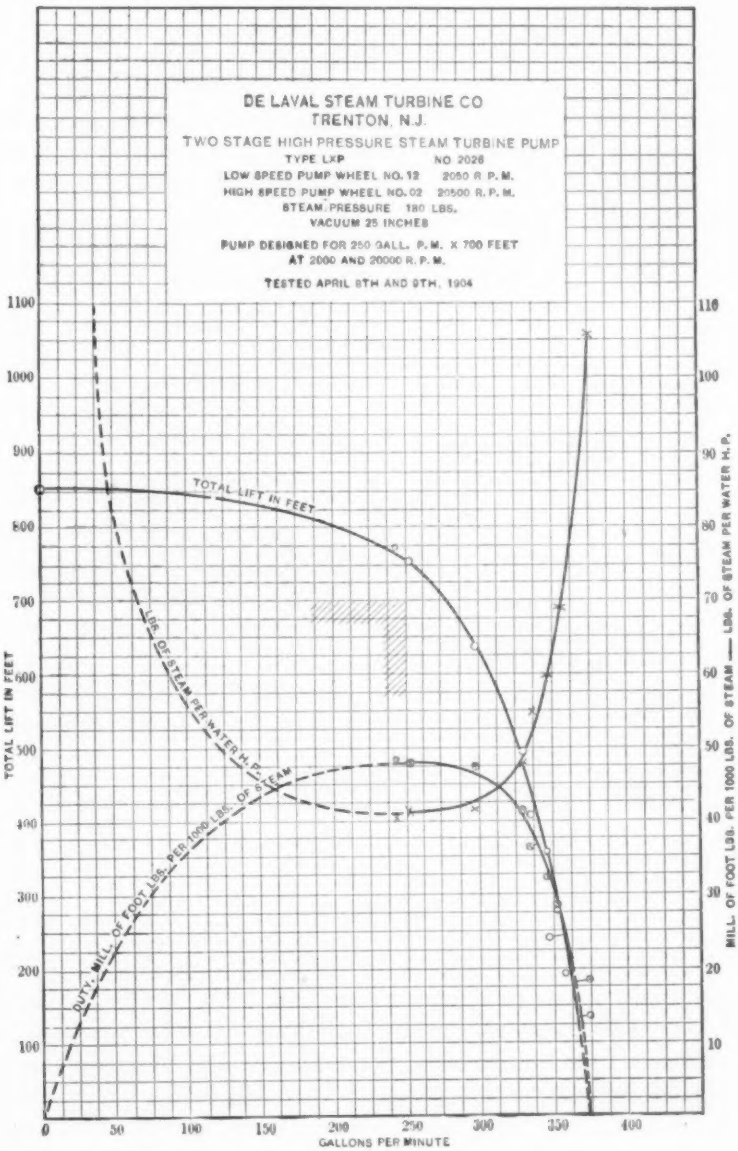


Fig. 513.—Tests by J. E. Denton and Wm. Kent.

22. Figs. 513 and 514 are curves from very exhaustive tests made recently by Professor Kent and Denton, on turbine pumps. These curves in themselves need no further explanation. The curve in Fig. 513 is obtained from a pump shown in Fig. 508. The curve in Fig. 514 from a pump of the type shown in Fig. 507.

23. In the foregoing, some frank statements have been made to illustrate the difficult theoretical as well as practical problems encountered, the solution of which have produced the present successful De Laval Steam Turbine.

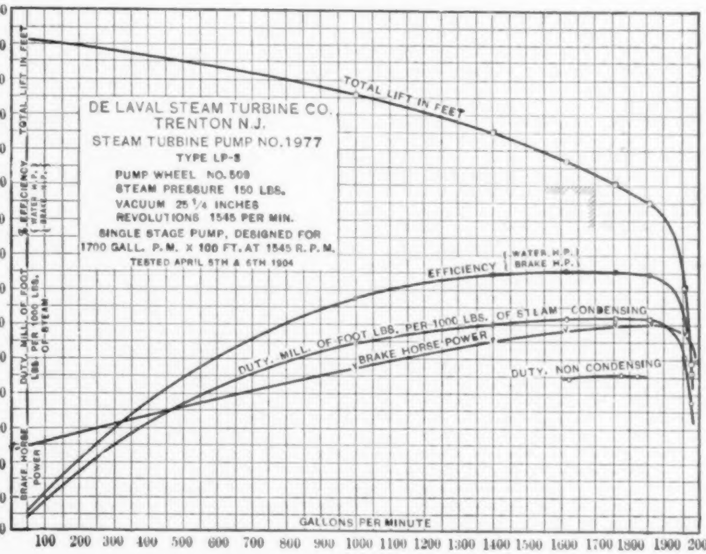


Fig. 114.—Tests by J. E. Denton and Wm. Kent.

DISCUSSION.*

Mr. Charles B. Rearick.—I do not really represent tne paper, but I want to make a remark or two that bears on the De Laval machine, and I wish also to say something about the question of condensers. On the question of condensers, I think the point of 26 inches vacuum being sufficient is very well taken, but it would seem that the amount of vacuum is for the engineer to determine. If he has a condition where fuel is very low in cost it follows that

* For further discussion on this paper, see the discussion appended to paper No. 1037.

he can afford to put in a plant at low cost and sacrifice economy. On the other hand, if his fuel is comparatively high he can easily afford to spend a little more money for his condenser. What is good in one case might not be good in another, and each case should be dealt with in reference to the conditions under which the machine will operate, as to cost of fuel, cost of getting water for the condenser, and other points that enter into the proposition. There is one point where the De Laval has made some progress, in which other turbines in the field, in the United States, have not done anything, and I expected some of the De Laval people would be here to say something in that connection. I regret that I have not any actual figures as to the number of De Laval machines driving pumps in actual service to-day, but there are a large number varying in size from 20 horse-power up to 225 horse-power, and I believe they are building several at the present time of 300 horse-power. They are particularly satisfactory for centrifugal pump work, which should interest engineers in general, since those pumps generally speaking are low in efficiency. The speeds of De Laval turbine driving shafts enable higher efficiencies for certain lifts than can be gotten at the lower speeds obtained from engines. Some of these efficiencies are showing up as high as 75 and 77, and some of the larger units that they are building are expected to give as high as 80 per cent. I have seen actual tests of 75 and 76 per cent. made by prominent engineers, Professor Denton and Professor Kent conducting them.

No. 1048.*

THE BURNING OF TOWN REFUSE,

WITH SPECIAL REFERENCE TO THE DESTRUCTORS AT BRUSSELS, WEST HARTLEPOOL, MOSS SIDE, AND WESTMINSTER.

BY GEORGE WATSON, LEEDS, ENGLAND.

Member, Institution of Mechanical Engineers.

1. *Refuse in Great Britain.*—Town refuse in Great Britain varies considerably in character according to local conditions, but it is a fairly safe generalization to say that it consists of one-third by weight of water, one-third combustible matter, and one-third incombustible. The last is withdrawn at the end of the burning process in the form of hard clinker.

The combustible material is largely vegetable and putrescible matter, but it includes, in the United Kingdom, varying quantities of unburned cinder from the wasteful open fires so dear to the English people in more senses than one. This cinder is much more plentiful in winter than in summer.

2. *Continental Refuse.*—Cinder is almost entirely absent from town refuse on the Continent of Europe, where closed stoves are used, producing a fine incombustible ash. The author has frequently proposed that the fine ash, which requires no treatment, should be collected in separate bins in each house, and not taken to the destructor at all.

3. *Collection.*—The nature of the material is affected largely by method of collection, which varies greatly in different towns. For instance, in Edinburgh there is a daily collection, the inhabitants being obliged to place their refuse on the street in receptacles which they provide themselves. These receptacles are usually anything but effective, and the contents are spread about by the wind and the rag-pickers, and mixed with the street sand; sometimes the same cart collects both street sweepings and refuse. The result is

* Presented at the Chicago meeting, May and June 1904, of the American Society of Mechanical Engineers, and forming part of Volume XXV. of the *Transactions*.

a very light sandy material, difficult to burn, which measures no less than 80 cubic feet to the ton.

In Bradford, Leeds and Sheffield, on the other hand, many large ash-pits are used, in some cases combined with privies, and they are only cleaned out at intervals of weeks or months. At one destructor in Bradford the refuse contains 40 per cent. of night-soil. Under such conditions the refuse is wet and heavy, measuring only 40 cubic feet to the ton.

In the author's opinion the prevailing custom of reckoning the amount of refuse burned by weight is misleading, and better comparisons as to the labor involved would be got by reckoning in cubic yards. It is obvious that a ton of bulky refuse requires much more labor in handling than a ton of such wet and heavy stuff as is collected at Bradford.

4. *Refuse as Fuel.*—Town muck is so heterogeneous that it is almost impossible to compare it with other fuels by any of the ordinary methods, and, in particular, calorimeter tests have always seemed to the author to be quite futile. It would appear impracticable to obtain a fair sample small enough to go into a calorimeter of a material comprising garbage, vegetable refuse, dust, straw, paper, rags, bones, broken glass, tin cans, wood, cinders, sacks, old boots, buckets, water-cans, casks, carpets and huge rolls of kamptulicon, to say nothing of the carcases of cattle, dogs, cats and pigs, the last being sometimes brought in numbers after an outbreak of swine fever. Amongst the articles gravely registered as having been destroyed at one of the early Leeds destructors was "one sea-serpent." In two instances parcels of explosives have found their way into furnaces, and blown out the fronts, fortunately without causing bodily injury in either case. To state the nature of the miscellaneous rubbish comprised in the refuse of towns is to demonstrate at once its unsuitability for agricultural manure, and the danger of allowing it to accumulate.

5. *Refuse as Manure.*—No farmer cares to cover his fields with broken glass and tin cans, and in the few instances, such as at Manchester and Glasgow, where some of the rubbish is taken by farmers, it has to be first sorted and ground, and then transported long distances free of charge by the Corporation at a cost exceeding that of destruction by fire.

6. *Danger to Health.*—Accumulations of this material involve the twofold danger of lowered vitality owing to an impure atmosphere, and direct propagation of disease through poisonous germs

carried by flies, rats, dust particles, running water and other agencies. Refuse heaps also frequently occasion great nuisance and expense by spontaneous combustion. "Destruction" of refuse is a somewhat scientific term, but it serves to distinguish the method of disposal by fire from the reduction or digestion processes used in the United States for recovery of grease and other matters, and from the slower process of nature, involved in disposal by dumping, which have been sometimes referred to as the "method of putrefaction."

The author does not propose to go through the history of destructors in Great Britain, where they originated. The late Mr. Alfred Fryer, of Nottingham, was the pioneer, and built his first successful destructor in 1875. It may be thought that nearly thirty years is a long time for the development which has taken place to have occupied. It must be remembered, however, that being dependent upon municipal enterprise, rapid advance has only been possible in recent years.

The author regrets to say that, in spite of the demonstrations made many years ago of the danger of filling old pits and hollows with this material, the practice of doing so, and of afterwards erecting dwelling houses upon it, has prevailed in most towns and cities in spite of all sanitary considerations, and has only given way to the use of destructors when every pit and hollow within the municipal boundary has been filled.

In passing over the history of the development of destructors, the author is also obliged to omit reference to many types which have proved successful, and have served to advance the general practice, and to many names honorably associated with such advance; and in dealing later with the details of certain installations has thought it best to confine himself strictly to his own experience. He will therefore deal only with plants on one particular system; at the same time remarking that there are many other destructors of different types in use which reflect the greatest credit on all concerned.

7. *Natural Draught.*—The earlier destructors were all upon the natural-draught system, with slow combustion, low temperatures, and little or no provision for raising steam. In charging and clinkering the furnaces large doors were kept open for about one-third of the time, and the strong chimney draught above the grates necessarily drew in tons of cold air over the fire through such openings, making high furnace-heats impossible. Frequent

complaints of nuisances from the chimney shafts caused by stinks and dust resulted.

8. *Fume Cremators.*—Fume cremators, introduced by Mr. Jones, the borough engineer of Ealing, were added in many cases, consisting of secondary fires in the flues fed with coal or coke, over which the products of combustion had to pass, thus scorching them and largely obviating the nuisance. These fume cremators certainly rendered a continuance of working possible in many cases, but owing to the high cost for fuel they were not always regularly worked.

9. *Maintenance.*—One advantage of natural draught and low temperatures was that the brickwork was easily maintained, and furnaces of the most ordinary construction have been found to last fifteen or sixteen years without much expense in renewing the fire-brick linings. Even to-day engineers are found who advocate low temperatures on this account.

10. *High Temperatures.*—Modern plants working at high temperatures with forced draught, and with steam-raising as one of the most important objects, involve much more difficult problems as regards construction and maintenance, and also as regards the protection of the stokers from heat and back-draught. No ironwork can last long in a modern destructor unless cooled in some special manner, and contrivances such as dampers within the furnace for closing or partly closing the draught outlet when the furnace door is opened, have become not only superfluous but impracticable. The author has learned to look with the greatest distrust upon any supposed improvement involving the introduction of unprotected ironwork into the hot furnaces or flues.

11. *Storage of Heat.*—In high-temperature furnaces heat is stored in the brickwork to such an extent that the furnace arch hardly ceases to glow even when nearly a ton of cold refuse has been freshly charged, and consequently the temperature does not fall below the point, say 1,250 degrees Fahr., at which septic gases might be given off.

12. *Plenum System.*—No one has contributed more to the attainment of such temperatures than Mr. William Horsfall, of Leeds, who first introduced the plenum system and the front exhaust flue. Recognizing that the principal cause of low temperatures in the old natural-draught furnaces was the inrush of a large excess of cold air over the fire during the lengthy processes of charging and clinkering, he set himself to apply forced draught

on the closed ash-pit system, first by means of fans and later by steam blast, and this enabled him to choke the furnace outlet in such a manner as to always maintain a slight pressure above the atmosphere within the furnace, thus absolutely preventing the admission of cold air when the furnace doors were open and allowing loose fitting doors to be used.

13. *Front Exhaust.*—He placed his exhaust flue in the front portion of the furnace arch, thus reversing the draught and bringing all the fumes given off by the refuse drying on the hearth over the hot fire and "cremating" them effectually within the furnace itself.

14. *Steam Raising.*—An important result of the introduction of high-temperature destructors has been that town refuse has now come to be regarded in some quarters as fuel. When it is remembered that the combustible matter in the charge is, say, only one-third by weight of the whole, the claims which have been put forward of a commercial evaporation of three pounds of water to high-pressure steam per pound of muck can only be regarded as ridiculous, for they would place the combustible portion of the material on a level with best Welsh coal. It is to be regretted that inventors have come forward from time to time offerings guarantees of such results in regular working. Their claims can only be accounted for by their lack of experience. Failure in the fulfilment of such guarantees under contract have led to a serious reaction, and there is now a tendency to discredit altogether the possibility of steam-raising from town refuse. As usual the truth is found between the two extremes.

15. *Steam as a By-Product.*—The great value of steam as a by-product of a destructor plant designed primarily for the disposal of the muck (but well arranged for the production and utilization of the steam) has been amply demonstrated. It may now be laid down with perfect confidence that, on an average, town refuse in Great Britain may be expected to evaporate in every-day working its own weight of water from and at 212 degrees Fahr. In many places it is safe to guarantee 1 pound of steam per 1 pound of refuse in summer and $1\frac{1}{4}$ pounds in winter. As much as 1.5 pounds may be got on test with careful management, as will be seen in the record of a test at West Hartlepool, given in Appendix 1.

The value of the refuse as fuel varies considerably in different districts, and according to the season and the amount of coal used

by each household. In some colliery districts the miners are provided with coal free of charge, and, as might be expected, the best steam-raising refuse is to be met with there.

On the other hand, the author has had experience of seaside towns, such as Lowestoft and Ramsgate, where in hot weather hardly any coal fires are found (gas-stoves being used for cooking); and the refuse consists largely of garden rubbish, garbage, and bad fish—poor stuff for steam-raising.

In wood-burning countries, and where closed stoves are used, not only does the absence of cinder mean a very poor fuel, but the fine ash (if collected with the refuse) has a tendency to surround and choke such combustibles as may be present, and to put out the fire altogether.

16. *Refuse Auto-Combustible.*—Nevertheless, the author has had, up to the present time, no experience of town refuse which was not auto-combustible, except that from the eastern district (the poorest part) of Berlin. Even there he has by no means given up the problem of burning the refuse by itself. Monte Carlo and Pernambuco are places where little or no cinder is found in the muck, and yet destructors are in successful operation at each of these without the addition of other fuel.

17. *High Pressures of Steam.*—To revert to the conditions prevailing in Great Britain, it used to be said that high pressures of steam could not be obtained from the heat of destructors; but a consideration of the temperature of the gases leaving the destructor, which varies from 1,700 degrees Fahr. to 2,000 degrees Fahr., might have shown the fallacy. As a matter of fact, steam is being obtained at Accrington (with a "Lancashire" boiler) and Moss Side, Manchester (with "Water-Tube" boilers), at 200 pounds per square inch, while pressures of 120 pounds per square inch are now quite common.

The conditions for the best working in the destruction of refuse are identical with those required for efficient steam-raising; and with high temperatures the clinker is harder, the working more rapid, and the chimney clearer of smoke.

18. *Arrangement of Cells and Boilers.*—The grouping of the cells and boilers is an important and much debated subject. Many early designers, attaching great importance to the radiations from the burning mass, held that the heating surfaces of the boiler should be immediately over the fire, and as close to it as possible. It is almost needless to say that the results were disappointing,

as the temperature necessary for complete combustion was never reached at all, owing to the cooling effect of the surfaces of the boiler.

Later developments of the same idea are still in evidence in the "sandwiching" of boilers and cells, that is, the placing of a cell on each side of a single boiler and delivering the products of combustion through side openings in the cells directly below the boiler tubes. This plan is much more satisfactory than the earlier one; but the chief drawback to its adoption is that when either of the cells is newly charged, and the gases may be escaping at a temperature below what is necessary for complete combustion, they may come into contact with the cool surfaces of the boiler unconsumed, and so pass away to the chimney, while fluctuations of steam pressure may also occur.

In the author's opinion, the best grouping of cells and boilers is to arrange the cells in blocks or batteries not exceeding six or eight in number, and to place the boiler or boilers as near as possible to the block of cells.

In the case of the back-to-back cells, arranged in such a block with the main flue underneath them, there is little or no loss by radiation, as the main flue is surrounded by cells. At the same time the immense advantage is secured that the products of combustion from all the cells are thoroughly mixed in the red-hot flue before passing to the boiler, and thus if one of the cells happens to be somewhat cooler than the rest, any unburned gases that may escape from it are thoroughly burned in the main flue, where there is always some excess of air. All the heat generated in the cells is carried forward by the hot gases, except the small proportion which escapes by radiation from the outer surfaces of the furnaces themselves.

With batteries of cells in a single row, Fig. 515, it is also possible to arrange that the loss by radiation shall be small. A further advantage is that the cells, being close together, the stokers have less floor space to work over, and the arrangement of railways or conveyors for removal of clinkers is simpler than with boilers and cells alternately.

19. *Working in Rotation.*—Under good management the working of the cells is kept strictly in rotation, a time-table and a clock being provided, and good time-keeping in charging and clinkering being insisted upon.

20. *Constant Pressure of Steam.*—With such methods the tem-

perature of the gases entering the boilers may be maintained almost constant, and the old complaint of the steam pressure

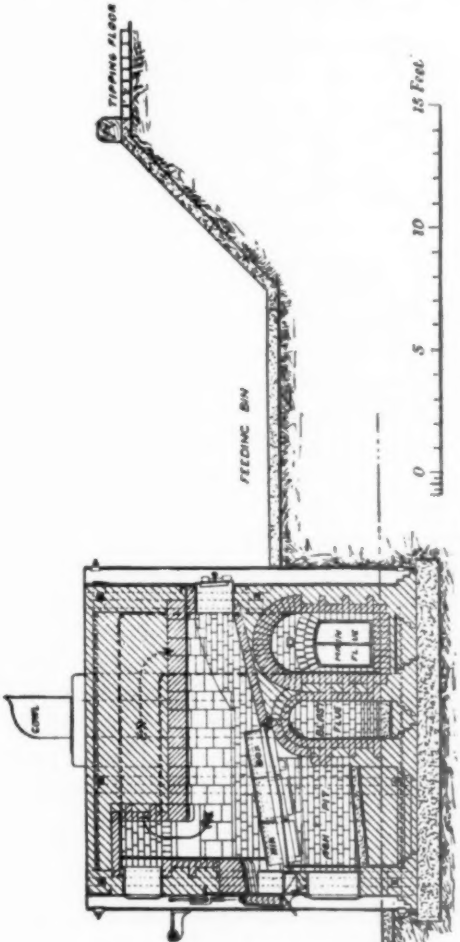


FIG. 515.—SINGLE ROW BACK-FED DESTRUCTOR FURNACE.

fluctuating disappears entirely. The clinkering and charging of each cell takes place every one and a half to two hours, according to the nature of the material. In some cases clinkering every hour

has been tried, but the experience of the author is that a longer run gives a harder clinker, and a better result all round, any small loss of steaming power being more than compensated by reduced labor and more perfect combustion. The capacity per cell depends mainly upon the grate area, and the strength of blast, but also, of course, very greatly upon the attention of the firemen.

21. *Blast.*—With refuse for fuel it is extraordinary how soon a high-pressure blast blows the fires into holes. With forced draught on the closed ash-pit system, a few minutes' use of the rake to fill up the blow-holes will increase the draught gauge from one-quarter inch to over an inch of water column. In some cases a draught of as much as 2 inches is used, but there are two great disadvantages to such pressures: first, that much more labor is required (almost constant trimming of the fires being necessary to fill the blow-holes), and, second, that the high blast sends up quantities of hot dust and sparks, which cake on the roof of the furnace and in the flues, necessitating constant cleaning.

22. *Grate Area.*—From 25 to 30 square feet of grate area has usually been provided in each cell up to recent years; but, of late, cells having 42 square feet of grate area have become common; and also much smaller cells are being introduced for special purposes, such as for use in hospitals, asylums and factories. For plants up to four cells furnaces of 30 square feet of grate area are most suitable; but for larger plants furnaces of 42 square feet are preferable, as the use of larger cells reduces the cost of labor. The length of the grate is usually 6 feet from back to front in both sizes of cells. It is customary to work throughout the twenty-four hours for six days a week; but in some cases, where steam is only required for night work, such as in electric lighting, the hours of working have been reduced, and the number of cells and boilers correspondingly increased. It is quite possible to bank the fires for twenty-four hours, and even longer, so that they do not require relighting on a Monday morning.

For the purpose of illustrating his remarks, the author has provided drawings and photographs showing several different types of plant in connection with the construction of which he has recently been employed. The installations chosen for this purpose are those at Brussels (twenty-four cells, Fig. 516), West Hartlepool (twelve cells, Fig. 517), Moss Side, Manchester (six cells, Fig. 518 and Fig. 519), and Westminster (six cells, Fig. 520 and Figs. 521 and 522). Each of these destructors is placed in a

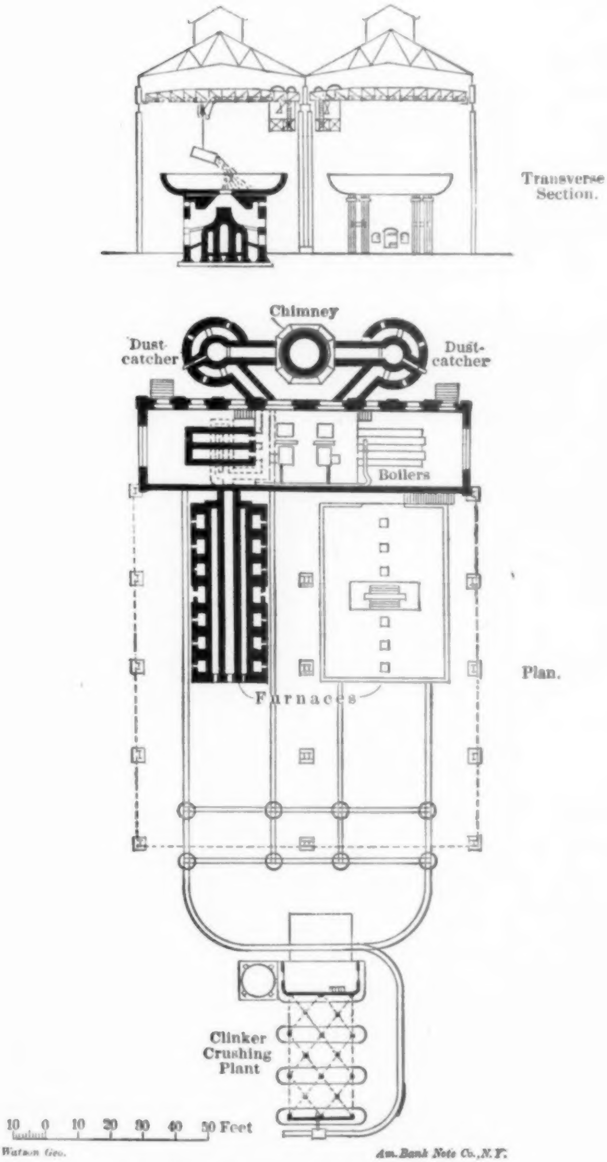


FIG. 516.—DESTRUCTOR OF 24 CELLS.

Grate area, 30 sq. feet each. City of Brussels.

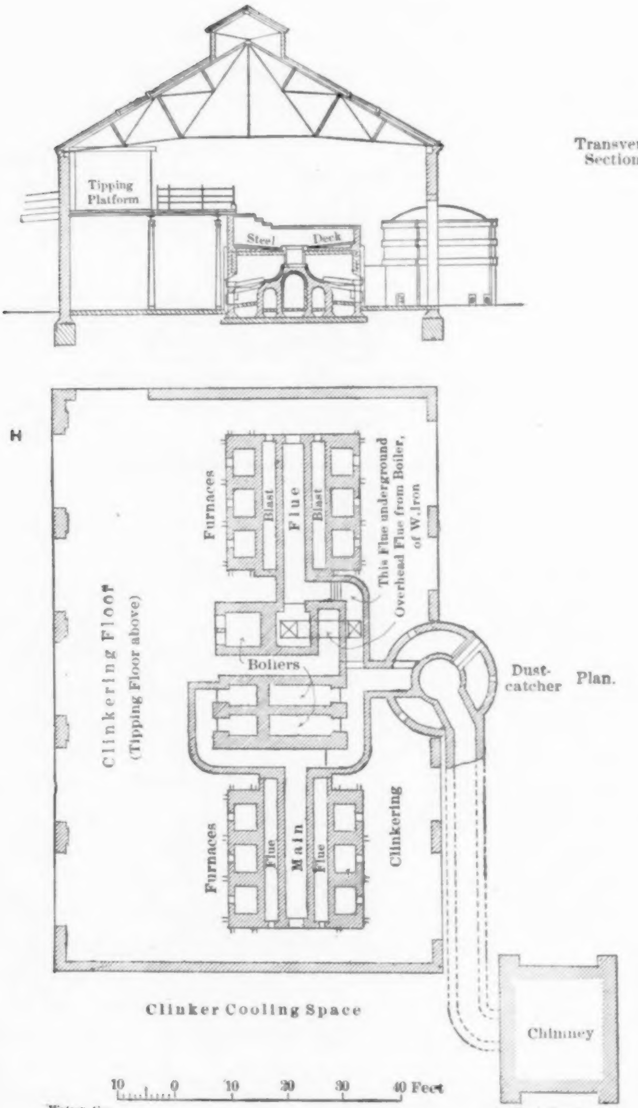


FIG. 517.—DESTRUCTOR OF 12 CELLS.
Grate area of 30 sq. ft. each. Corporation of West Hartlepool.

densely populated district, and all are in full operation without causing any inconvenience or complaint.

23. *Destructors at Brussels and West Hartlepool.*—The Brussels and West Hartlepool furnaces, Figs. 516 and 517, are of the back-to-back type, fed by hand through holes in the deck on the top of the furnaces. These feed holes are of somewhat peculiar construc-

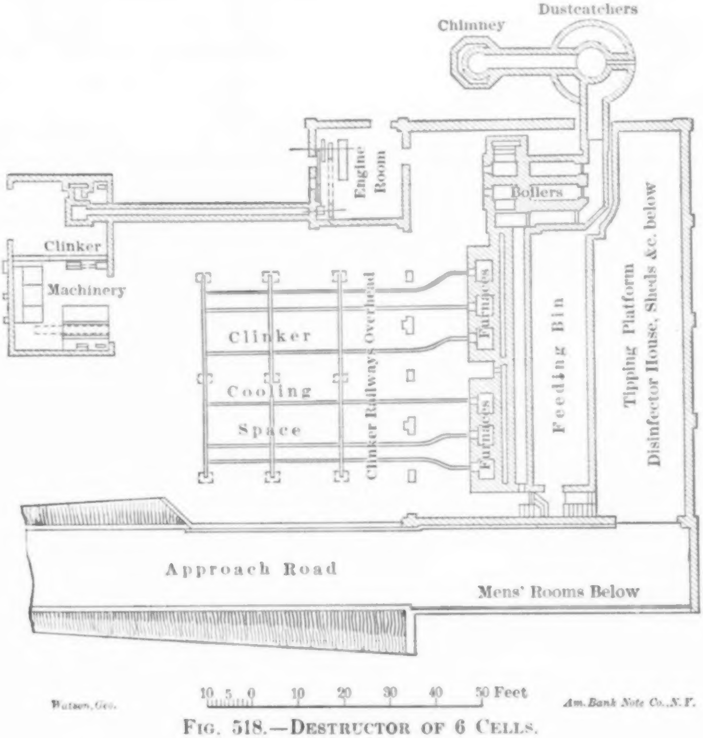


FIG. 518.—DESTRUCTOR OF 6 CELLS.
Grate area of 30 sq. ft. Moss Side, Manchester.

tion: they are so designed that no lid is required, the refuse itself being trodden down into the holes to stop them up. It will readily be seen that the use of a lid on a deck littered with rubbish would be very inconvenient. In order that the method of stopping above referred to may be used, there is provided immediately below the feed-hole (which is common to two furnaces) a flat table or saddle so arranged that when refuse is dragged or shovelled into the hole and allowed to lie on the saddle it fills up and chokes the opening. When it is desired to charge either furnace the refuse

is simply pushed off the saddle with a prong—a very simple operation, requiring very little labor.

The deck is 6 feet below the level of the tipping floor, thus forming a convenient bunker for the reception and storage of the muck as near as possible to the feed-holes. In the West Hartlepool plant the deck is formed of steel plate separated from the top of the furnace by an air space in order that it may be kept cool; while at Brussels the deck is carried entirely by the furnaces and is formed of reinforced concrete. At West Hartlepool the refuse is carted up an inclined way with a gradient of one in twenty,

FIG. 519.—FURNACES. 6-CELL PLANT. MOSS SIDE.

while at Brussels there are two electric overhead travelling cranes, one over each group of twelve cells, which lift the cart-bodies off the wheels and dump their contents on to the decks. The latter system has also been adopted at Hamburg, and in the plant now being completed at Zürich.

24. *Moss Side Destructor.*—At Moss Side, as will be seen on reference to the drawing, Fig. 518 and Fig. 519, the six cells are arranged side by side in a single battery with the two water-tube boilers at the end of the battery, the charging openings are at the back of the furnaces, and are provided with suitable furnace doors. These doors are now generally planed to fit their frames, and are lined with fire-bricks. They are raised and lowered by levers and balance weights, and give a good command of the furnaces, which slope downwards to the front.

The refuse is tipped from the carts into a bunker of concrete, and having a sloping back, so that the toe of the heap of refuse is

never far from the surface mouth, into which it is thrown by a shovel. This method would seem at first sight to involve more labor than the top feeding, but it is found in practice that, while it costs more to put in the muck with a shovel, the labor in front of the furnace in dragging forward and trimming is correspond-

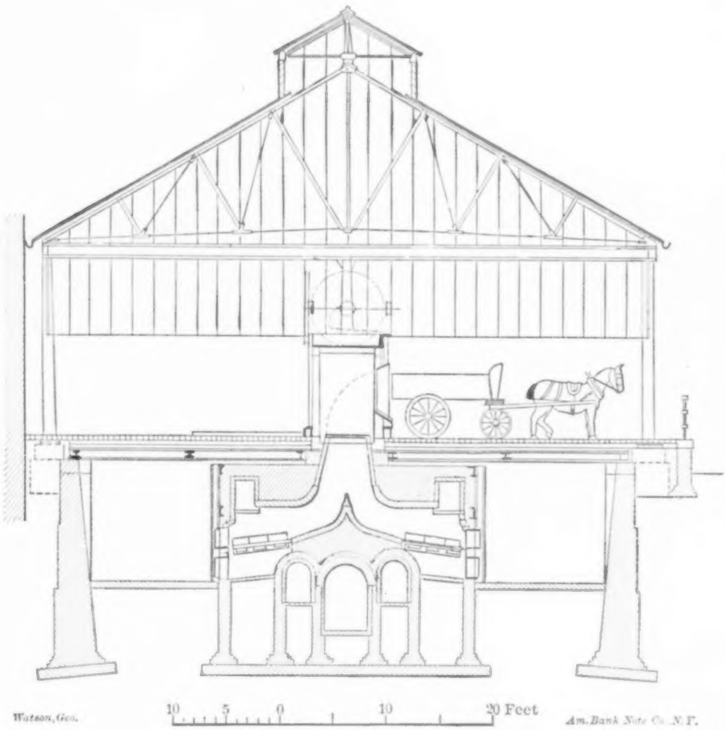


FIG. 520.—DESTRUCTOR OF 6 CELLS.
Grate area of 42 sq. ft. each. City of Westminster.

ingly lessened. It may be remarked, however, that even at West Hartlepool, where the refuse is not bulky, the labor of charging costs one-third of the stokers' wages, and it is, therefore, a matter of the utmost importance to reduce it as much as possible.

25. *Westminster Destructor.*—The Westminster Destructor, shown in Fig. 520 and Figs. 521 and 522, has been erected on the Shot Tower Wharf, at the south side of Waterloo Bridge, in the center of the metropolis, and serves a district of which Covent Garden Market and the Strand form important features. The

refuse is exceedingly bulky, consisting largely of market and shop refuse, paper, packings, straw, and all kinds of light material, measuring not less than 90 cubic feet to the ton. To feed such stuff by means of a shovel would be very costly, and therefore special means were demanded. The conditions have been met by a new design of furnace, into which the four-wheeled collecting carts and steam motor-wagons tip their loads direct, no storage being provided other than the spare carts which are brought in from the yard and discharged as required. Sufficient spare wagons are provided to keep the destructor in full operation day and night. The furnaces are six in number, each having 42 square feet of

FIG. 521.—WESTMINSTER DESTRUCTOR. EXTERIOR.

grate area. They are arranged back to back, with a water-tube boiler at the end of the battery. A very large feed hole, 6 feet by 4 feet, is common to two furnaces, there being thus three holes in all.

The carts can be brought to either side of the feed-holes to dump. Great difficulty was experienced at first in arriving at the correct shape for the feed-holes—that is, expanding downwards, as now shown, so that the refuse cannot arch over in the furnace mouth. A worse difficulty was that of getting such large lids as were required to be quite smoketight when closed. Metal to metal joints and grooves filled with sand were in turn found unsatisfactory, and the handling of such heavy lids was also difficult. Eventually hinged lids, counterbalanced by weights hung from volute quadrants, so as to be perfectly balanced at all positions, were adopted.

Each lid dips into a water-seal which surrounds the furnace mouth, the water, which evaporates slowly, being maintained at the proper level by a ball cock. This makes a perfectly gas-tight joint, and not only is the trouble of smoke from the feed-holes obviated, but it is found that the absolute tightness of the joint, preventing even a small leakage, causes the furnace gases to pond up beneath the lid and to remain stagnant, thus forming a shield of comparatively cool gas, which protects the lids from the furnace

FIG. 522.—CART TIPPING, WESTMINSTER.

heat. As long as there was any leakage at all, the heated gases drew that way, and kept the furnace mouths and lids red hot, which caused them to crack and give trouble.

A hopper, constructed of wrought-iron plates and set on hinges in a position at right angles to that of the lid, is lowered over the opening, so as to prevent refuse getting into the water-seal. This hopper is balanced in a similar manner to that described for the lids, and both hopper and lids are easily worked by hand by means of large chain wheels. The labor of charging is thus saved, the muck is never handled at all, but shot straight into the furnaces. This is an advantage from a sanitary point of view, but it is not universally appreciated, as it puts an effective stop to trade in rags, bones, and dirty glass bottles, which can no longer be picked out by the stokers and sold.

Before leaving the subject of the Westminster destructor, which is the first of its kind, the author wishes to acknowledge his great indebtedness to the Engineers of the City, Mr. Bradley and Mr. Ventris, and to Dr. Priestly, the Medical Officer of the Health of Lambeth (whose duty it was to inspect the working of the plant) for their invaluable aid in overcoming the initial difficulties of the undertaking.

The author has already remarked that he would like to see the duty of destructors reckoned in cubic yards rather than tons, as giving a fairer basis of comparison. It is not, however, in the least likely to come into vogue, since the wagon-weighing machine is such a simple and convenient method of taking the quantities.

26. *Labor in Working.*—The author would, however, suggest a further reform in comparative figures to the effect that the labor bill in working the furnaces should be reckoned, not in pence or shillings per ton (it varies even in England from sevenpence half-penny at Moss Side and elsewhere to two shillings and threepence at several London destructors, for furnacemen only), but in tons dealt with per man per hour. This would eliminate the discrepancies due to the stokers' wages and hours of labor varying so much in different districts. A destructor in which a stoker can deal with a ton per hour during an eight-hour shift may be considered satisfactory, and this is being attained at Westminster. There can be no doubt that a very large saving is effected in charging direct from the carts in the case of a destructor dealing with 72 tons of bulky refuse, such as that of the Strand district, every twenty-four hours.

27. *Mechanical Stokers.*—Many attempts have been made to effect the operations of stoking and clinkering by mechanical grates, but hitherto without much success. Mechanical grates of many kinds have been tried, and large sums sunk in such experiments. The author has had some expensive experience in this direction, and the opinion that he has formed is that any attempt to make the burning process continuous instead of intermittent will fail, unless some entirely fresh method be found. The quality of the refuse as fuel is too poor to enable the fire to creep back through a comparatively thin layer of it as fast as the material must be moved forward to give anything like a reasonable output; and after the mechanism has been set so as to give only about three of four tons per cell per twenty-four hours, it has been found that the speed was too great for the fire, which was very soon

all ejected from the furnace. The fuel also varies so much that one part of the fire will have become black clinker while other parts are insufficiently burned. Trimming and clinkering by hand and the intermittent system of firing in one-and-a-half or two-hour heats at present hold the field.

28. *Clinkering.*—The operation of clinkering is one requiring both strength and skill. The mass of clinker is often 5 inches thick over the whole grate surface, and in breaking it up and withdrawing it from the furnace the stoker must be careful to turn it over so as to throw off any live fire on the top, which he afterwards spreads evenly over the grate for the purpose of lighting the new charge of refuse. The blast, which is shut off during clinkering, should be put on again for a few minutes before charging, so as to prepare a bed of hot fire for the reception of the charge.

Another method adopted with some well-known types of furnace is to feed them continuously from the front by hand and to clinker also continuously from each portion of the grate surface in turn. In furnaces both fed and clinkered at the front, no drying hearth is provided, and there is therefore no preliminary drying of the refuse. This system is adopted by the author for small portable destructors and furnaces for hospitals, but not for larger installations, on account of the higher cost of labor. Some engineers use continuous grates without any division into cells above the grates, the ash-pits only being divided. The author, however, strongly favors the cellular system, which lends itself much more readily to repairs, as one cell can be repaired at a time without stopping the others, thus rendering a duplication of plant quite unnecessary.

29. *Detailed Description of Furnace.*—The different methods of charging the furnaces having been described, it may now be convenient to deal with the details of those parts of the furnaces which are common to the three types at Westminster, West Hartlepool, Brussels and Moss Side respectively. The refuse is first piled up on the drying hearth above the main flue and is afterwards raked from the front on to the grate-bars as may be required. The drying hearth and grate-bars slope down about one in six towards the furnace front; this greatly reduces the labor of working the furnaces. The grate-bars have narrow spaces—about three-sixteenths of an inch—and are made in single lengths of 6 feet, four bars being cast together. This enables the chisel tools used in clinkering to be worked from below along the whole length

of the grate-bars without check. A wide "deadplate" is provided in front, and the furnace mouth is closed by a door extending the full width of the furnace and lined with fire-brick blocks. The furnace door slides upwards to open, the working faces being planed, and the door suspended from a balanced lever. In closing it falls into wedge-shaped catches which force it tight against its frame. Such doors are found much more suitable than hinged doors, which were commonly used in early destructors, as the latter exposed their red-hot inner surfaces to the stoker on being opened.

In each main furnace door is fitted a small rake-door, just large enough to admit the stoker's rake to enable him to trim the fire and pull down the muck without exposing himself to the heat of the furnace. The main door need only be opened for the operation of clinkering. The ash-pits are closed by suitable air-tight doors, and are sloped every way towards the door, to facilitate the removal of ashes.

30. *Fireclays.*—The whole of the interior of the furnace is constructed of specially made fireclay blocks of the best quality, dovetailed together, and all of the arched construction. Flat fireclay lumps over openings and flues are found to crack. Owing to the great heats attained and the frequent changes of temperature, it is necessary to select the fireclay with care, to avoid troubles from expansion and contraction. Bricks too rich in silica expand and contract far too much, and clays which answer well for metallurgical purposes are often found too brittle for destructor work. The fireclays which are found suitable for use in destructor furnaces neither contract nor expand under heat, and are able to withstand frequent changes of temperature, the maximum being about 2,300 degrees Fahr. They should consist of from 60 to 70 per cent. silica and 30 to 40 per cent. alumina. A slight admixture of iron is harmless, but lime, potash and soda, which are usually found in such clays, should only be present in very small quantities, as they tend to act as a flux. The beds of fireclay at Leeds, Sheffield, Glenboig, near Glasgow, and Stourbridge are all of suitable quality. The bricks and blocks must be very truly formed, in order that the thinnest possible joints may be made. It is also important that the fire-bricks should be a little thicker than the common bricks with which they have to bond, because naturally the joints in the common brickwork will be thicker than would be desirable in the fire-brick work.

31. *Fans or Steam-Blast.*—The draught is forced by means of

fans or steam-jets. The question as to which gives best results, dry air or steam-jet blast, is frequently debated. A blast of about one inch of water column is found to answer best with average refuse. The use of higher pressures causes the fire to burn into holes too frequently, and thus causes too great an excess of air to pass, and consequently lowers the temperature.

This pressure can be obtained easily and economically either by steam-jet blast or by a centrifugal fan. For working at a rate of 10 tons per twenty-four hours, on a grate area of 30 square feet, or at a rate, say, 30 pounds of refuse per square foot of grate per hour, a volume of about 700 cubic feet of air per minute at atmospheric pressure and temperature for each cell is required, or, say, 23 cubic feet of air per minute for every square foot or grate area in use. To deliver this quantity of air at the requisite pressure requires, with an efficient steam-blast apparatus, about 100 pounds of steam per hour for each cell of 30 square feet grate area, whereas with a good centrifugal fan only one-fifth of this amount, or, say, 20 pounds of steam per hour, is needed. To compensate for the larger consumption of the steam-jets, it may be stated that they give a higher temperature and evaporation than the dry-air blast, provided that the refuse is rich enough in carbon to give the necessary temperature for the dissociation of the steam, which forms water-gas in the furnace, and which greatly improves the combustion. This action has been explained * by Lord Kelvin and Dr. Archibald Barr in the following words: "A more important function is, however, fulfilled by the steam. In coming into contact with the incandescent fuel it is decomposed, the hydrogen being freed, while the oxygen combines with the carbon in the fuel to form carbon monoxide. This decomposition of the water is effected by heat abstracted from the lower part of the fire, where it can be of comparatively small value for the cremation of the distillate. The 'water-gas' (hydrogen and carbon monoxide) passes upwards to be burned by the excess air which it meets with over the fire, thus serving to increase the temperature, which would otherwise exist at the meeting of the products of combustion with the gases distilled from the raw material."

The importance of the action of the steam-jet was strikingly exemplified in an instance which came under the observation of the author at Bury, Lancashire, where in a new destructor it was not

* Report on the "Horsfall Destructors," 1898.

found possible to reach the guaranteed temperatures in the flues by means of dry-air blast. The substitution of steam-blast for the fans immediately gave and maintained the required temperature. It is fully demonstrated that, provided always the refuse is sufficiently rich in carbon, a steam-blast, although it uses more steam for the draught, yet so increases the amount of the total steam raised as to give a better steaming result on the whole than the fans. With the refuse of Hamburg and Berlin, however, it is doubtful whether the same can be said, because the use of closed stoves in those cities renders the refuse very poor as regards combustible cinders. One hundred pounds of steam used in the blast for each cell of 30 square feet grate area per hour may be considered a very good result for steam-blast, and it can only be secured by using efficient arrangements.

A steam-jet has been devised by Mr. C. W. James, in conjunction with the author, for this purpose, in which the nozzle is flat instead of round, thus yielding a ribbon of steam instead of a plug of steam. It is probable that the air is carried through the blast-tube mainly by means of surface friction between the jet of steam and the surrounding column of air, and it is therefore advantageous to have the greatest possible surface on the steam-jet per pound of steam passing. This is, of course, obtained by making the jet flat and thin. A thickness of one-fortieth of an inch is found best. It is also important that the steam should be superheated, so as to prevent condensation and obstruction in the nozzle itself. A pressure of 30 to 40 pounds to the square inch for the steam-blast gives the most economical results. Reducing-valves are used, giving this pressure whatever may be the working pressure of the boiler.

A further great advantage to be set down to the credit of the steam-blast, as compared with fan draught, is the fact that it protects and lengthens the life of the ironwork exposed to heat in the furnaces, such as, for instance, grate bars and the cast-iron side-boxes which are used for heating the blast. After leaving the nozzle the steam in the blast is condensed, and it reaches these iron parts in the form of particles of water. On striking the hot iron it is immediately re-evaporated into steam, carrying with it a considerable amount of heat, and thus keeping down the temperature of the ironwork. The author has seen grate bars taken out of a furnace after six years' constant use very little the worse for wear, whereas with dry air-blast and high temperatures the grate

bars would require to be renewed after six months to twelve months at the latest.

32. *Cast-iron Furnace Sides.*—The steam-jet trumpets used in the West Hartlepool installation are combined with the cast-iron side-boxes which form the sides of the furnaces above and below the grate-bars. These side-boxes serve the double purpose of heating the blast to a temperature of 400 degrees Fahr. in the ash-pit and of protecting the brick-work at the sides of the furnace from the undermining action of the hot clinker. In furnaces with brick-work sides, at the first level it is found that the clinker fuses to the brick-work, and every time it is removed brings away particles of the wall, thus gradually undermining it, and allowing the crown of the furnace to fall before it is anything like worn out. The side-boxes, which are kept cool by the passage of the cold air and by the cooling action of the steam above described, prevent this action. The air is drawn into the boxes from hoods, something like the hood over a blacksmith's fire, placed over the clinker-door in such a position as to draw off any smoke and dust rising during the operation of clinkering. This improves the ventilation of the stoke-hole.

33. *Firing Tools.*—The firing tools consist of prongs or pushers for charging the furnaces, light and heavy rakes, for pulling down and clinkering respectively, and chisel bars. They are all necessarily long, and any means of making them lighter, without reducing their strength, is worth adopting. It is found a good plan to use a weldless steel tube for rake handles, and to make the chisel bars of solid masons' tool steel in order to get the greatest possible strength.

34. *Boilers.*—In the early destructor plants multitubular boilers of plain cylindrical shape, with fire-tubes about 3 inches or 4 inches in diameter, were frequently used. One of the most serious drawbacks to these was the fact that the dust carried in the gases soon choked up the tubes, which had to be brushed out three or four times in the course of a day, and an even worse defect was the stiffness of the boiler, which did not easily yield to expansion and contraction. A destructor boiler, placed at the end of the main flue, is, of course, really a gas-fired boiler, and it is subject to having the gases switched on or off instantaneously.

It was found that with these multitubular boilers the tubes very soon began to leak at the tube plate; moreover, boilers of such a large diameter as were generally used were not very suitable for

high pressures, unless they were constructed in a very expensive manner. For these reasons such boilers are now only used for very small installations, or where steam-raising is a matter of little importance.

It is essential that a boiler which permits its tubes to expand and contract very freely should be used.

The two types of boiler most favored at the present time in connection with destructors are the Lancashire type and the water-tube type. The former has the advantage of large capacity for storage of hot water and steam, and it is also less susceptible to troubles arising from hard water, but it is found, when using destructor gases, that the flue area is insufficient unless the boiler be made larger than would be usually adopted for the same rate of evaporation when fired in the ordinary way with coal. A deposit of dust in the two flues of the boiler is not a very serious drawback, as the boiler is easily cleaned, and the lower half of the flue is, of course, less valuable as heating surface than the upper half. A well-constructed Lancashire boiler is also able to withstand the effects of expansion and contraction, due to the hot gases being suddenly switched on or off by means of the by-pass damper.

In cases where the destructor has to work continuously, while the power is only required for, say, three or four hours a day (so often in the case in electricity stations) the large capacity of the Lancashire boiler for storage of steam power is very valuable.

The water-tube boiler, however, seems destined to have the preference in most new destructor installations. In the first place, it is specially adapted for high pressures.

Steam is raised very quickly, which is a great advantage in starting the destructor at the beginning of each week, because the forced draught cannot be put into operation until steam has been raised. The water-tube boiler is well adapted for resisting the sudden changes of temperature to which it is subjected in a destructor, and it provides ample area for the passage of the gases in proportion to its heating surface. It has another great advantage in the fact that dust can only lodge on the tops of the tubes (when in proportion to its heating surface. It has another great advantage in the fact that dust can only lodge on the tops of the tubes (when the tubes are horizontal or sloping), leaving the lower surfaces, which are, of course, the most effective, always clean. The dust can be readily removed by means of steam-jets applied through suitable cleaning holes, provided in the brick-work sides of the boiler seating.

When water-tube boilers with vertical, or nearly vertical, tubes are used, dust cannot lodge at all, and as there is no soot in destructor gases, an externally clean heating surface is always presented. It may also be remarked that boilers of the water-tube type can be arranged closer to the end of the battery of destructor cells, and the damper arrangements can be made more convenient than when boilers of the Lancashire type are used.

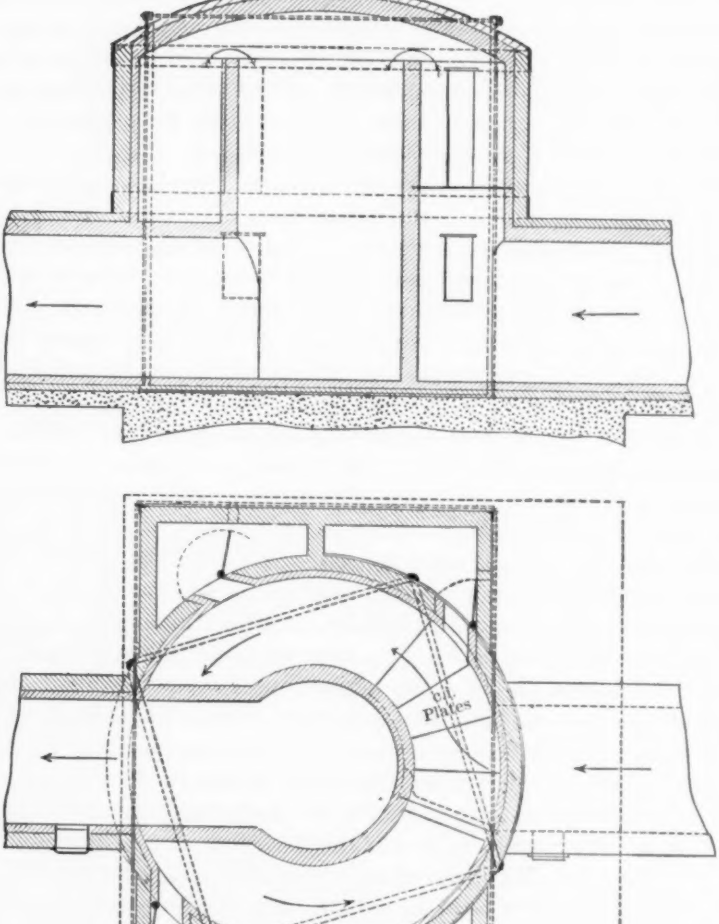
35. *Fuel Economizers.*—Seeing that the destructor depends much more upon forced draught than upon the pull of the chimney, it is economical to carry the reduction of the temperature of the gases at the chimney base to a low point, say between 400 degrees and 500 degrees Fahr., and fuel economizers are frequently placed in the flues behind the boilers. These should be, like the boilers, arranged so that the gases can be by-passed at short notice.

36. *Utilization of Heat.*—The heat of the gases from destructors, although, as stated above, of considerable value, is unfortunately often lost through lack of convenient application near to the site of the destructor. It is a curious fact that, although electric-lighting stations only demand a considerable quantity of power during three or four hours per diem, the combination of destructors with electricity stations is the commonest method of utilizing the heat. There are, however, other kinds of municipal work that can absorb a fair proportion of the power available. Stone-breaking, crushing and screening the clinker from the destructors, or grinding it into mortar, driving repairing shops, sawing, chaff-cutting, sewage and water-pumping, heating baths and wash-houses, and even schools and dwellings, have been carried out in different places by means of the steam from destructors.

At Moss Side (Manchester) a snow melting pit has been constructed by the Municipal Engineer—Mr. Longley—in which a series of pipes bringing steam from the destructor boilers is placed. These are perforated so as to throw jets of hot steam upon the snow as it is tipped into the melting pit.

At Folkestone, the Borough Engineer—Mr. Nichols—has constructed heating pans on the flues beyond the economizer for the purpose of warming the materials for use in forming streets of tar Macadam.

37. *Dust-catcher.*—A dust-catcher has been placed between the boiler and the chimney in the installations at Brussels, West Hartlepool, Moss Side, and in many other recent destructors. The



Watson, Gen. Am. Bank Note Co., N.Y.

FIG. 523.—DUST-CATCHER.
5-Cell Destructor for Blackpool.

latest form of this apparatus has been devised by Messrs. Newton and Diggle, of Accrington, in conjunction with the author, and it consists of a swirling chamber, into which the gases are led from the main flue in such a manner as to give them a circular or revolving motion within the chamber, Fig. 523. In carrying this out they are made to pass round an annulus formed by the outer circular wall and the inner well, built up to within a short distance of the domed or arched roof. The whole of the interior is lined with fire-brick to enable the apparatus to deal with red-hot gases when the boilers are not in use. The rapid revolution of the gases within this chamber throws all the suspended dust—even the smallest particles—against the outside wall of the chamber by centrifugal force.

At intervals, vertical slits are provided, leading into pockets outside the chamber.

The dust, travelling along the outer wall, finds its way into these slits and thence into the pockets, where the gases are stagnant, and where the dust falls. The vertical slits leading into these pockets can be closed by suitable doors hung freely on ball-bearings and so arranged that whenever the outer cleaning door of the pocket is opened for the purpose of withdrawing the dust the door covering the inner opening is closed by suction.

With this apparatus it is possible to extract the dust from any or all of the pockets, without interfering with the draught of the destructor. This is important when destructors are combined with electricity stations, as at Accrington, because it is never practicable to allow the flues to get cool enough to clean out by hand. Not only the dust-catcher itself, but the whole of the flues throughout the destructor and boiler can be cleaned by simply opening the end doors, and allowing the wind to rush through, whipping up the whole of the dust deposited in the flues, and carrying it forward into the dust-catcher, where the centrifugal action discharges it into the pockets above described. It will be observed that the stronger the draught, the more complete is the separation of the dust.

38. *Dampers.*—Dampers of many kinds, including cast-iron (with and without ribs), wrought-iron and steel plates of different thicknesses have been tried. Cast-iron gas-valves, with cold-water circulation, have also been extensively used. The latter were successful until—as always happened sooner or later—the water supply was interrupted, by accident or design, when they would imme-

diately fail. Experience seems to point to a damper formed of fire-brick blocks in a suitably designed cast-iron frame as the most reliable for general purposes. The damper should be set in grooves deep enough to protect the frame from the action of hot gases. It should also be provided with suitable means for making the slit at the top practically air-tight, when the damper is either closed or open.

The author has recently adopted double by-pass dampers for the by-pass to the boilers. These are found to be practically gas-tight, as the damper nearest to the chimney so reduces the tension of the draught that the leakage past the second damper, with which the gases come earliest in contact, is negligible.

39. *Chimneys.*—A dissertation on destructor chimney shafts would be out of place in a paper which has already run to considerable length, but it may be remarked in passing that, unless in very special circumstances, a destructor chimney need not exceed from 100 feet to 120 feet in height. Even at the Westminster destructor, which is, as before stated, in the heart of London, the height of the chimney above ground line is only 90 feet, and the destructor is worked without complaint. In fact, it may be said that it is difficult to tell by observation of the chimney whether the plant is at work or not.

Destructor chimneys should be constructed to withstand the full heat of the gases, which are sometimes directed into them without passing through either boilers or economizers. For this reason they should invariably be lined to the top with fire-brick, and it is good practice also to provide an air space, properly ventilated, between the fire-brick lining and the outer shell.

40. *Clinker-handling.*—To come now to some of the machinery accessory to destructors, it may be mentioned that one of the most important points is the handling of the hot clinker as it is withdrawn from the furnace. At Moss Side, and also at Westminster, overhead railways, on the system devised by Mr. Cox, City Engineer of Bradford, and the late Mr. McTaggart, are used. These railways consist of a single H beam suitably suspended, on which a small trolley, with ball or roller bearings, runs freely, Fig. 519 and Fig. 524. From the trolley hangs a large tipping truck of suitable capacity for taking the whole of the clinker from the cleaning of a single cell. This represents two, and even sometimes three, very large barrow-loads, which would be heavy labor to wheel. The clinker truck is suspended immediately under the lip of the front

dead-plate, so that the clinker can be withdrawn directly into it. It is then pulled away by the stoker along the runway over the clinker cooling bed, and tipped on its own trunnions so as to discharge the clinker on the ground, where it is allowed to cool. These railways are capable of considerable modification to suit different sites. For instance, at Westminster, where the plant is sunk below ground level, a section of the rail is lifted on a

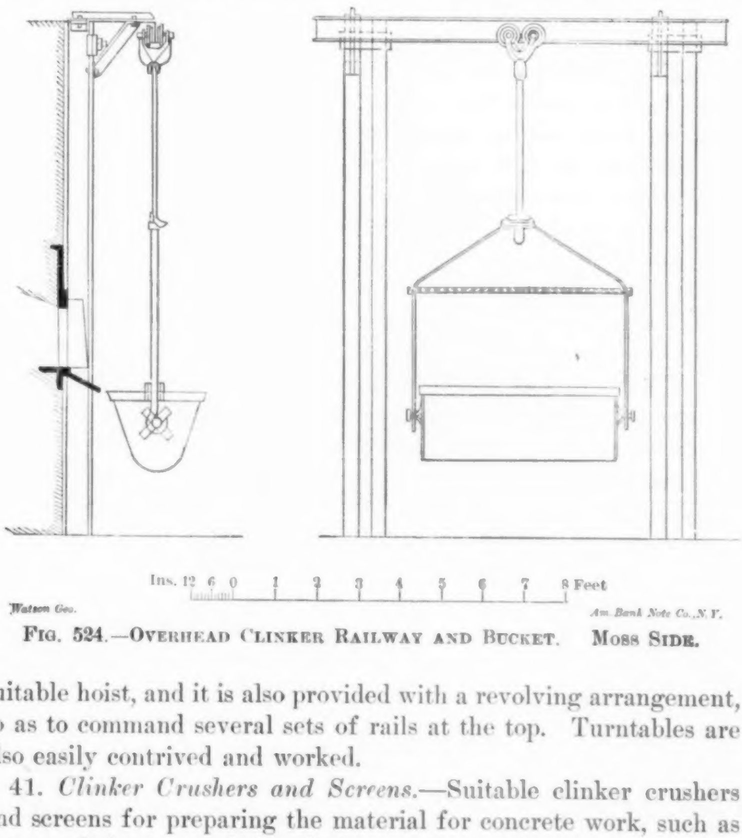


FIG. 524.—OVERHEAD CLINKER RAILWAY AND BUCKET. MOSS SIDE.

suitable hoist, and it is also provided with a revolving arrangement, so as to command several sets of rails at the top. Turntables are also easily contrived and worked.

41. *Clinker Crushers and Screens.*—Suitable clinker crushers and screens for preparing the material for concrete work, such as fire-proof floors, road foundations, and the like, have been designed also by Messrs. Cox and McTaggart, and are extensively in use. The crushers consist of a pair of fluted rollers mounted on wooden keys so as to give the necessary elasticity to avoid breakage when pieces of iron are encountered. The yield of the keys is sufficient to stop the machine gradually, and not instantaneously, and has the effect of throwing off the belt of the machine, and

avoiding the sudden jar which would cause breakage. The screens deliver the material in sizes required for the various works to be undertaken.

42. *Paving Flags.*—Attention has recently been given also to the production of concrete flags for footways, and to mixing and pressing machinery for making building-bricks.

43. *Clinker Bricks.*—Dr. Schultess, of Zürich, has worked out a very complete system for the production of bricks from clinker and lime. One of the features of his system is the lime-slaking machine, which produces a perfect lime in a dry powder, free from nodules, and slaked with exactly the proper amount of water. He also provides mixing machinery arranged so as to automatically mix the correct quantities of ground clinker, lime and water, and one of the advantages to which he draws special attention is the fact that no water need be expressed from the flag under compressure, the amount of moisture supplied being what is required for the proper setting and no more. This enables a brick or flag with very clean arrises to be produced. A further feature of Dr. Schultess's system is the maturing of the final product in forty-eight hours by means of steam at atmospheric pressure.

Bricks and flags so produced have a strength considerably exceeding that of good burned clay bricks, and the bricks are suitable for foundation work, embankment walls, and for inner walls of houses. Their dark color is a drawback to their adoption for outer walls. In the ordinary way Portland cement is used for such productions, but Dr. Schultess claims to get a better result with lime, and, of course, at a lower cost.

44. *Clinker Mortar.*—Mortar made from clinker and lime is found to be almost equal to hydraulic mortar, and it is capable of withstanding considerable heat. In fact, in destructors at Oldham, Bradford and elsewhere it is frequently used for pointing up the interior of the furnaces, and is found to stand as well as fire-clay.

45. *Destructors for Hospitals, etc.*—Destructors of small size, suitable for villages and public institutions, such as hospitals or large hotels, have been alluded to above. No special remark is called for respecting the design of such furnaces, except that they should be of the simplest character. As a rule they are both fired and clinkered at the front through the clinkering-door. In other respects, such as in the details of grate bars, cast-iron side-boxes, forced-draught apparatus, front-flue openings, and so forth, the

design of the larger types of furnaces is followed. In come cases, however, where it is impossible to get steam for steam-blast or electric current for fan draught, such furnaces have to be worked by means of strong natural draught.

46. *Portable Destructors.*—" Portable " destructors are also made, consisting of a furnace following the above general lines, attached to a plain cylindrical " multitubular " boiler with a dust-catcher in the smoke-box, steam-jet forced draught and a short chimney; the whole being mounted on wheels for transport. The

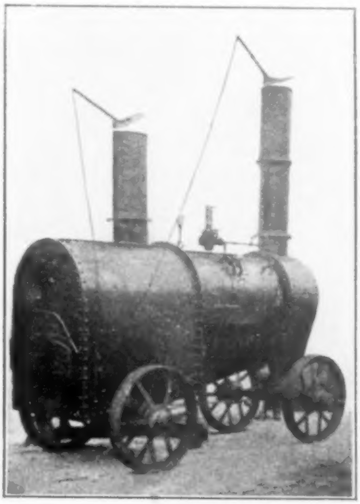


FIG. 525.—PORTABLE DESTRUCTOR.

weight of the destructor illustrated, Fig. 525, is between 6 and 7 tons, and the capacity about 4 tons of muck in the 24 hours. These small " portable " destructors are good steam-raisers, and when used in connection with hospitals the power may be conveniently utilized in connection with electric lighting plant, Röntgen ray apparatus, disinfecting chambers, and the like. They are also intended to be worked when stationary.

A still smaller destructor has been devised by the author's brother, Mr. F. L. Watson, for military purposes, consisting of a small furnace combined with sterilizing tanks, in which infected clothing, etc., can be dealt with, the whole being mounted on two wheels and suitable for mule or horse transport. There can be no doubt that a convenient apparatus of this kind, if available at all

standing and advanced camps, would materially lessen the ravages of zymotic diseases in time of war.

In conclusion, the author begs to acknowledge his indebtedness to many engineers, in not a few instances friends of his own, for their labors in connection with the disposal of refuse by fire, and he trusts that the above notes of his own experience may be of interest and of service to the members of the Institution.

APPENDIX I.

THE DESTRUCTOR AT WEST HARTLEPOOL.

TEST REPORT.

NOTE.—The following figures and the accompanying diagram, Fig. 526, give the mean results of the two days.

Date of test	28th and 29th Jan., 1904.
Duration of test	48 hours.
Number and type of cells	6 back-to-back.
Total grate surface	180 square feet.
System of forced draught	Steam jets.
Nature of refuse	Ashpit, nightsoil, market.
Number of firemen and average wage per day	9 at 5 shillings.
Number, size, and type of boilers	1 Babcock and Wilcox, 2,393 square feet.
	Tons. cwt. qr. lbs.
Total quantity of refuse burned	272,432 lbs. = 121 12 1 20
" " " per cell per 24 hours	22,703 lbs. = 10 2 2 23
" " " per square foot of grate per hour	31.5.
Tons per man per shift	6.7 tons.
Cost of labor per ton burned	8.9 pence.
Total water evaporated	348,673 lbs.
" " " per hour	7,264 lbs.
" " " per square foot of heating surface per hour	3.03 lbs.
" " " per lb. of refuse from and at 212 deg. Fahr. or 100 deg. C.	1.56 lbs.
Mean steam pressure	155 lbs. per square inch.
" feed temperature	43 degrees Fahrenheit.
" main flue temperature	Above 2,000 deg. Fahr. (beyond range of Pyrometer).
" temperature behind boiler	534 degrees Fahrenheit.
Horse-power developed at 20 lbs. steam per indicated horse-power per hour	363.
Purpose for which steam is utilized	Electric lighting

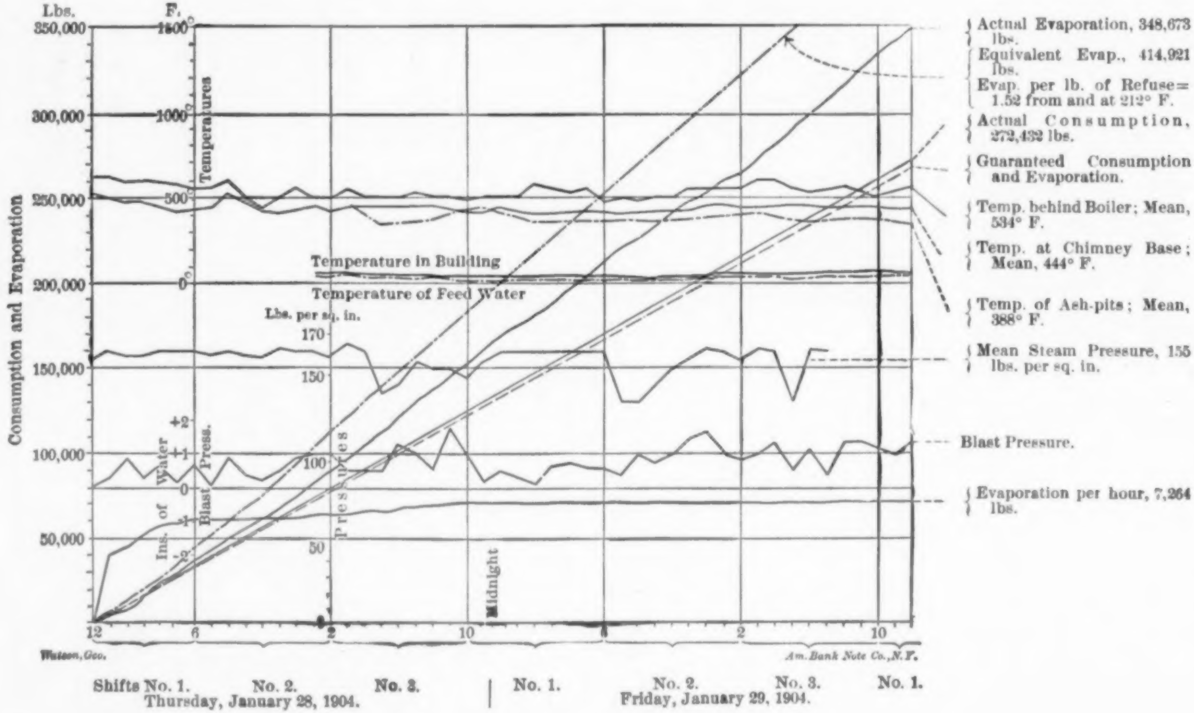


FIG. 526.—CHART OF TEST RESULTS, WEST HARTLEPOOL.
Hourly Observations. Temperature in Main Flue beyond Pyrometer reading.

APPENDIX II.

CITY OF WESTMINSTER.

Test Report.

Date of test	2d Dec., 1902, to 4th Dec., 1902.
Duration of test	$45\frac{1}{4}$ hours.
Number and type of cells	6 back-to-back direct cart-fed.
Total grate surface	252 square feet.
System of forced draught	Steam jets.
Nature of refuse	House, trade, and market.
Number of firemen and average wage per day	9 stokers at 35 shil. per week. 4 top men at 27s. 6d. per week.
Number, size, and type of boilers	1 water-tube, 1,426 sq. ft. H. S.
	Tons. cwt. qrs. lbs.
Total quantity of refuse burned	138 15 1 0 = 310,828 lbs.
" " " " " per cell per 24 hrs.	12 5 1 8 = 27,476 lbs.
" " " " " per square feet of grate per hour	— — — — 27.2 lbs.
Tons per man per shift	8 3 2 5
Cost of labor per ton burned	11.5 pence, exclusive of engineman and foreman.
Total water evaporated	Test for evaporation abandoned, boiler blowing off very heavily.
" " " per hour	
" " " per square foot of heating surface per hour	
" " " per lb. of refuse from and at 212° F. or 100° C.	
Percentage of clinker and ash to refuse burned	24.9 per cent.
Mean steam pressure	125 lbs. per square inch.
" feed temperature	48 degrees Fahrenheit.
" main flue temperature	Well over 2,000 degrees Fahr. softened mild steel.
" temperature behind boilers	500 degrees Fahrenheit.
Purpose for which steam is utilized	Electric light, steam jets, hoist engines, etc.

DISCUSSION.

*Mr. J. Hartley Wicksteed.**—I know Mr. Watson very well, and I may say that he has been extremely successful with Mr. Horsfall's destructor. Destructors are very difficult things to undertake, because the town refuse varies so much in quality in the different towns, and it requires a very great deal of experience to hit off a result that you can guarantee. The original basis which differentiated Mr. Horsfall's destructor from others was the use of the steam jet. The action of the steam jet is very

* President of the Institution of Mechanical Engineers.

abstruse, but it has been reported upon by Lord Kelvin and Dr. Barr, and I will presently read to you one paragraph—not that I wish to direct your discussion to that particular point, but because it was what started this company making destructors. Of course, experience has shown that there are a hundred considerations which are of quite equal importance to the original idea concerning the utility of the steam jet, but still you must have something to start with, and Mr. Watson's company had this to start with and they worked at it and it lead them into the knowledge of all the other points. The thing itself I have, no doubt, is of considerable value. The author says in his paper, "it may be stated that they (that is, the steam jets) give a higher temperature and evaporation than the dry-air blast, provided that the refuse is rich enough in carbon to give the necessary temperature for the dissociation of the steam, which forms water-gas in the furnace, and which greatly improves the combustion.

This action has been explained by Lord Kelvin and Dr. Archibald Barr in the following words:

"A more important function is, however, fulfilled by the steam. In coming into contact with the incandescent fuel it is decomposed, the hydrogen being freed while the oxygen combines with the carbon in the fuel to form carbon monoxide. This decomposition of the water is effected by heat abstracted from the lower part of the fire, where it can be of comparatively small value for the cremation of the distillate. The water-gas (hydrogen and carbon monoxide) passes upwards to be burned by the excess air which it meets with over the fire, thus serving to increase the temperature which would otherwise exist at the meeting of the products of combustion with the gases distilled from the raw material."

So that it seems the chief function of the steam jet is to shift the locality of the intensest heat of the fire.

This paper and Mr. Russell's paper were prepared at the request of the American Society, and we from England have put before you the best knowledge possessed by any members of our Institution, and I hope you will take the matter up for discussion.

*Mr. Alfred Saxon.**—Sanitary engineering up to the present has had no particular attractions for me; in fact I have rather looked upon the science as a necessary evil, but seeing that the paper as presented seemed to my mind to be representative of the engineering of Leeds and its district, I thought I would see

* Member of the Institution of Mechanical Engineers.

whether no good thing could come from Manchester, which I represent, and, as a matter of fact, I believe we have the best destructor for the purpose produced in the Manchester district.

There are some points of agreement on this question, notably in respect to the quality of the refuse. I find that the people who have spoken and written upon this question agree with the statement of the author that an average for this is one-third by weight of water, one-third combustible matter, and one-third incombustible. On other points, of course, there are differences of opinion. The claim that Mr. Watson makes, for the isolated cell system at West Hartlepool works, does not, as a matter of fact, seem to be sustained, nor does it produce results which will compare with the continuous grate system which is adopted by the Manchester firm. The claim also made by the author about the ease of repairs in the isolated cells is doubtful, when one cell is adjacent to another, working at 2,000 degrees Fahrenheit, is it not rather too much to expect that much good work will be done in the cells which are supposed to be isolated, but which must to a certain extent be heated up by the cells which are in work.

Now, I should like to make a comparison of the test which is given in the paper with one which was made by the National Boiler Insurance Co. at Woolwich on the Destructor erected by Messrs. Meldrum Bros., Timperley. At Woolwich on three grates, 75 square feet, 63¼ tons of refuse were burned in 24 hours, and each pound of refuse evaporated 1.9 of water. Whereas at Hartlepool, in the test given in the paper, on 180 square feet, they burned 121 tons. In the one case, at Woolwich, at 77 pounds; whilst in the other case only 31 pounds per square foot.

Mr. Edward N. Trump.—While the papers presented by our English friends on the above subject are of great interest as showing engineering skill in handling a difficult problem, from the point of view of the chemical engineer the very name "destructor" suggests a process which should only be a last resort in the *utilization* of our wastes.

The remains of the beefsteak for which you paid two dollars a pound, and failed to eat because of a bad appetite; the butter left on your plate; and trimmings from the raw materials; which from such a large percentage of the food which is finally consumed, all contain valuable ingredients which should be utilized not destroyed.

The work in the United States has been more in the direction of the "reduction" of "garbage" or kitchen waste, to its useful

constituents rather than its destruction. The fermentation which so quickly starts up in these waste, producing a crop of germs very dangerous to the health of the community may be stopped, and the waste thoroughly sterilized by boiling under high pressure, and the valuable greases and liquids saved and utilized.

First, The waste must be gathered as frequently as possible to reduce the fermentation to a minimum. Second, It must be collected in covered carts or wagons, kept as air-tight as possible to prevent offensive odors escaping along the streets. Third, A number of large steel tanks or digestors of a strength sufficient to allow an internal steam pressure of one hundred pounds per square inch receive the garbage, forming an excellent storage for a fluctating delivery, and it is thoroughly cooked with direct steam until all of the dangerous germs are killed, the greases extracted and the whole thoroughly digested. Fourth, The products of this digestion are now utilized as follows: The contents of the digestors are emptied out in continuous filter, which separates the fibre from the liquid, pressing out as much of the latter as possible. The fibre is dried and may be burned in a special producer or furnace, with the production of large quantities of useful gas and of ammonia. The liquids are evaporated to get rid of the surplus water after the valuable greases are allowed to separate, and the resulting sirupy liquid mixed with the ashes from the fibre form valuable fertilizing. The greases are refined and often utilized in the production of toilets soaps. Many of the best known soap makers are using these greases from the reduction works.

Several kinds of machinery are already in use in this country to produce the above results. It only needs engineering skill to produce valuable products from the waste, and there is more than sufficient carbon in the dried fibre to make the heat and steam needed to operate the plant.

It would, therefore, appear that the *reduction* of waste rather than its destruction is the economical process to develop, and incineration should only be resorted to for those dry wastes which may be readily utilized as fuel, and which will stand storage without fermentation.

*Mr. George Watson.**—The author thanks the president for his very kind remarks. It is quite correct that Mr. Horsfall started his destructor business with the steam jet for forcing the draught. The steam jet, as pointed out somewhat briefly in the paper, has

* Author's closure, under the Rules.

several advantages, among which it may be again mentioned, that it protects the grate bars and other iron parts of the furnace from becoming red hot, while at the same time in many cases it increases the temperature for the combustion of the refuse. But it should not be supposed that the " Horsfall " system necessarily involves the use of steam jets.

In many cases it is found preferable to use fans, and care should always be taken to consider the exact conditions to be met in making the selection.

In a recent instance (at Guernsey) both steam jets and a fan have been provided in a " Horsfall " destructor.

This arrangement can now be made with little increase of cost over the fan system, and it is one which offers obvious advantages.

The remarks of Mr. A. Saxon serve to illustrate the difficulty in which the author was placed in preparing his paper, and which he endeavored to meet by confining his remarks strictly to his own experience. He also tried to avoid preferences as far as possible, and in indicating the advantages of the " Cellular " type of destructor, he expressed his own opinion, while endeavoring to avoid undue bias. Moreover, this is just one of the points constantly debated upon which the author considered himself free to speak his mind, for the reason that the " Cellular " system is no monopoly of any one firm, but is free to be adopted by all, as Mr. Saxon well knows. The author is not aware which of the claims put forward for the West Hartlepool Works is the one which Mr. Saxon considers is not sustained. He can only say that he is quite unaware of any inaccuracy in his statement. He must also say that in his opinion the results obtained at West Hartlepool are comparable to those obtained on either of the " Continuous-Grate " systems at present on the market, and he leaves members to make their own comparison; having regard to steam raising; not only on trial, but in every-day working; to cost of labor per ton burned; to durability; and to ease of repair. As to the expectation of being able to repair one cell while the others are at work which Mr. Saxon thinks unwarranted, the author can only say that under his own supervision such repairs are frequently made without difficulty.

While congratulating Mr. Saxon on the results obtained under test at Woolwich, the author may be allowed to say that there is no difficulty in burning a large amout of refuse per square foot

of grate per hour, either in the "Continuous-Grate" destructor, or in the "Cellular" destructor.

It is a question simply of strength of blast, and of frequency of raking over and of clinkering. In other words, the main consideration is that of the labor involved in the operation. It is the author's experience that much less labor is involved in burning at a rate of 30 pounds per square foot of grate per hour than at 70, while the clinker produced is harder, and the process is more complete, clinkering every two hours, than when clinkering at more frequent intervals.

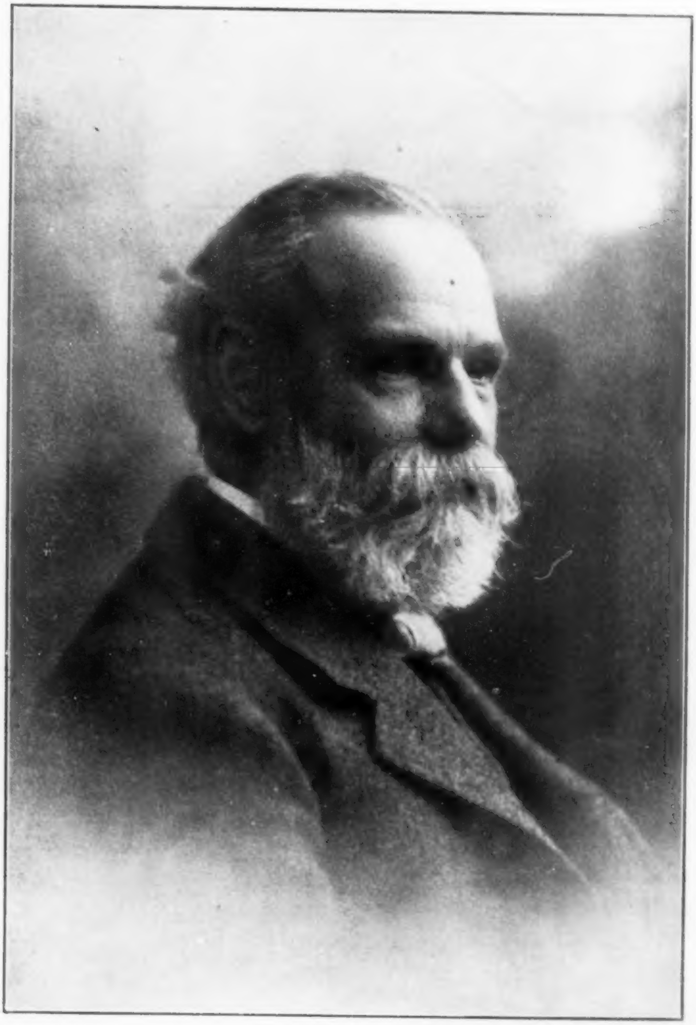
In regard to the remarks of Mr. Trump, the author supposes it is more attractive to everybody to utilize than to destroy. But the engineer must divest himself of sentiment and reduce the question of reduction versus destruction to figures.

The impression gained in England as to the system of reduction prevailing in the United States is that, generally speaking, they do not pay, or at least that they will not continue to pay, and further it is practically impossible to conduct the process without creating a nuisance.

Such plants are often the subject of legal injunction, and where they escape the reason seems to be that they are placed at a considerable distance from dwellings.

In making these somewhat sweeping statements the author is painfully conscious of his lack of experience, but he is merely reflecting the general opinion which is held in Europe, which may be far from correct.

He does not understand how the digesters can form an equalizing storage for a fluctuating delivery, unless they can be discharged while under pressure, which he believes is not the case. Otherwise they must be completely charged and then sealed up till the process is complete, and in the meantime the refuse continually arriving must be otherwise stored. The author's own belief is that refuse, even in the United States, is constantly getting less valuable for any purpose, even for steam raising, as sanitary science progresses, and that if one-half of the skill employed to utilize the grease from the refuse were applied to the utilization of the clinker from the destructor, better results all round could be obtained from a centrally situated plant without nuisance, and with a large saving in cartage to put to the credit of the destructor.



ROBERT HENRY THURSTON.

No. 1049.

ROBERT HENRY THURSTON.

IN MEMORIAM.

For the fifth time in the history of the Society it is called on to record the death of an engineer who has held its Presidential office. The present occasion reflects particular interest from the fact that Professor Thurston was the first President of the Society, and was called to office on its organization, serving for two years. The other presidents who have preceded him in the necrological list are Messrs. George H. Babcock, E. B. Coxe, E. F. C. Davis, and J. F. Holloway.

Dr. Thurston was born in Providence, October 25, 1839. It adds a distinct note of the unusual in his death that it should have occurred on the evening of his sixty-fourth birthday, on October 25, 1903. He was preparing to greet a group of friends convened to celebrate the day, when suddenly, and so far as known painlessly, he was seized with an attack of the heart and passed away, without regaining consciousness, within a very short time.

Professor Thurston's activities naturally group themselves under three departments. He was eminent as an engineer and practitioner, as an educator and investigator, and as an author. He inherited his mechanical talents and instincts by direct descent. His father was Robert Lawton Thurston, who was born at Portsmouth, R. I., in 1800, and died at Providence, R. I., in 1873. Robert L. Thurston, as partner with John Babcock in 1834, formed the Providence Steam Engine Company, the first steam engine building establishment in New England, there being only two others bearing earlier date in the United States. In 1838, by the withdrawal of Mr. Babcock, it became Robert L. Thurston & Company, and in 1845, by other changes, the firm became Thurston, Greene & Company. The large machine shop of the old firm still remains, forming part of the works now occupied,

in 1904, by the Rice & Sargent Company, building steam engines. The firm of Thurston, Greene & Company, bought from Sickles the patent covering the principle of lifting the spindle of a steam engine valve by a mechanical means, and then releasing or tripping the catch which lifted the valve, so that it closed quickly by either gravity or a spring.

This firm seems to have been the first manufacturers which built a standard form of engine working with the principle of expansion. Thurston, Greene & Company were succeeded by Thurston, Gardner & Co., who were the plaintiffs in the suit with Mr. George H. Corliss for an infringement of the Sickles patent, and after several decisions favorable to the plaintiff these were reversed, on a final appeal based on the contention that a valve spindle was not necessarily the same thing as a valve stem. It was during this period of litigation that Mr. Greene invented the type of release gear, which has since been known by his name.

Robert H. Thurston, who was the oldest son of Robert L., inherited his mechanical instinct, and during his childhood and youth spent much time in his father's shop. He graduated from Brown University, in Providence, in 1859, with the degree of Ph.B. and C.E. Ten years later he received the degree of A.M., and in 1899 the degree of LL.D., when his fame was won. For two years after graduation, or until the breaking out of the Civil War of 1861, he was engaged with the firm of which his father was the senior partner.

On the breaking out of the war, his tastes led him to enter the Engineer Corps of the United States Navy, and he served at the front, under Dupont and Dahlgren, until the close of hostilities. When second assistant he was put in charge of the "Chippewa," as Chief Engineer in 1863; was made 1st-Assistant, in 1864, and transferred to the iron-clad "Dictator." He was present at the battle of Port Royal and at the siege of Charleston. At the close of the war in 1865, he was detailed to the United States Naval Academy at Annapolis for duty in the department, which was then called the Department of Natural and Experimental Philosophy.

On the death of the professor at the head of the department, Dr. A. W. Smith, he was placed in charge under Admiral David D. Porter, who was Superintendent of the Academy at that time. He was made chief engineer in 1866, and remained in this relation until, in 1870, he was called by President Henry Morton to the

Chair of Mechanical Engineering then created for the first time in the Stevens Institute of Technology of Hoboken, N. J. From this time on, while also pursuing professional work, Dr. Thurston became principally an educator and investigator. His professional work after accepting this position included his service as a member of the International Bureau of the Vienna Exposition in 1873, which was made the opportunity of extensive visits to metallurgical establishments in England, Belgium, and France. His service on the Bureau resulted in the securing of some valuable American awards to engine builders and exhibitors of machine tools. He served on a board appointed by the federal government to investigate the underlying causes of boiler explosions and was the reporter for this board. What are known as the Sandy Hook tests of that testing board were largely planned by Professor Thurston. Shortly thereafter (1875) he became a member of the United States Testing Board associated with Messrs. Holley, Sooy-Smith, Laidley, David Smith, and Beardslee. He was Secretary of this Board, and its leader, preparing details, writing official papers and carrying the laboring end of the work. It was a great disappointment to him when the Board's life expired by limitation and it was thought inexpedient to continue it.

It was in connection with this work that Professor Thurston's attention was turned to the investigation of alloys with which his name is identified. During all the rest of his life he also maintained an active relation to practice as a Consulting Engineer and expert, although both of these functions were dwarfed by his work as an educator and author.

When Mr. Edwin A. Stevens made his original bequest of $600,000 to endow an institution of learning under trustees, it was decided that this institution of learning should be an institute of technology. By the advice of President Henry Morton, who was summoned to be the first president of that institution, it was further decided to specialize in the direction of Mechanical Engineering. It was to the genius and wisdom of Professor Thurston that the technical education of this country and in the world owes its debt for recognizing the importance of the laboratory of engineering as a prime factor in the success of the education of the mechanical type of technical man. It is difficult for him who looks back on thirty years of successful development along this line to appreciate the magnitude of the problem before Professor Thurston who had the conception and development of

the laboratory on his hands without the guiding of any experience whatever. The conditions in Europe were not useful for the purpose in hand; the atmosphere of the employer of technical men was unfavorable to a college training of young engineers and the financial problem of equipping a laboratory, which is great at any time, was made nearly impossible to the man who had the beginnings to make when standards of all sorts were very different from what they are at the present time, when there were fewer wealthy manufacturers and no group of successful technical men to constitute an argument in discussion. It was easier for Professor Thurston to begin the mechanical laboratory at the point of testing materials so that his earlier reputation was founded upon his special achievement in this line. He devised the Thurston autographic testing machine and the installation of a Riehle machine in the institute laboratories antedates by a few months the construction of the first Fairbanks testing machine in the laboratories of Columbia University. From that time on, the laboratory conception of the Stevens Institute grew steadily in importance and meaning, and by the interest of friends and well-wishers, the equipment in machine tools kept pace with the developments of its reputation in other directions.

He remained at Stevens Institute until 1885. These were years of intense activity in his work as author and investigator, as well as teacher and expert. In 1873 he was United States Commisioner for the World's Fair at Vienna, and his report and letters are full of revelations of his assiduity and acumen. Professor Thurston secured as one of the consequences of his labor a liberal proportion of the awards for American exhibitors. In 1875 the special work on alloys with which Professor Thurston's name is identified, was in progress, and the results are to be found in the reports of the United States Board for the year 1878. In 1876 his health failed seriously from nervous exhaustion, and, although in 1878 he resumed his work with partially restored health, it was not until October of 1880 that he resumed his work at the Institute completely cured.

It was during these years from 1880 to 1882 that he served the Society of Mechanical Engineers as its first President, and in co-operation with Holley, Worthington, Sweet and others, underwent the severe and exacting duties of presiding over the early work of organization and starting of the infant society. He presided at its first meeting in November, 1880, and took upon him-

self the preparation of presidential addresses, both for that meeting and for the subsequent one on his re-election to a second term.

He always remained in most earnest and enthusiastic relations of service to the Society, as a past-president, attending its Council meetings when possible, and always participating in the discussions.

Stevens Institute conferred on him the degree of Doctor of Engineering, on the severance of his relations in June, 1885, to take the work of Director of Sibley College of Mechanical Arts at Cornell University in Ithaca. For eighteen years, from 1885 to 1903, Dr. Thurston's great work was the building up of Sibley College.

The number of students at the beginning was comparatively insignificant but had become in the neighborhood of 900 under his enthusiastic and wise leadership. The courses grew to include both post-graduate and special work, and the equipment from a valuation of about $100,000 to over half a million in buildings and contents. In addition to this he was much sought as a contributor to the engineering organizations of which he was a member and for service on important technical commissions. The following is a somewhat incomplete list of the services which he rendered in this direction:

At Stevens Institute and at Cornell; the United States Testing Board, consisting of Messrs. A. L. Holley, William Sooy-Smith, Colonel T. S. Laidley, U. S. A.; Chief Engineer David Smith, and Commander L. A. Beardslee; the United States Boiler Test Commission. On both of these he served as Secretary and to a notable extent the leader. He was a member of the New Jersey State Committee to report a plan for encouragement of manufacturers of ornamental and textile fabrics, a member of the United States Commission of safe and vault construction for the United States Treasury, of the New York State Commissions to report on a modern rifle for the National Guard, and also to examine and authorize voting machines.

He was Vice-President of the American Institute of Mining Engineers in 1878, Vice-President of the American Association for the Advancement of Science at Nashville, in 1877, in the absence of Professor Pickering, elected at the preceding meeting, was regularly elected to serve in 1878 at the St. Louis meeting of the Association, and in 1884, at Philadelphia, in which year he was honorary vice-president of the British Association for the

Advancement of Science. He was the first President of the American Society of Mechanical Engineers, 1880. He was member of the International Juries of Vienna, 1873, Paris, 1889, and Chicago, 1893. On the occasion of the visit of the American Engineers to Europe in 1889, he made the address in response to the address of welcome by the President of the British Institution of Civil Engineers in London, and was one of the speakers at the banquet of the British Institution of Mechanical Engineers held in Paris in July, 1889. He was officer de l'Instruction publique de France, an editor (for engineering) of Science, of Johnson's Cyclopædia and Century Dictionary.

He was connected with the following and other societies: American Association for the Advancement of Science; American Institute of New York; American Society of Civil Engineers; American Society of Mechanical Engineers; American Institute of Mining Engineers; Army and Navy Club; British Institution of Naval Architects; Royal Institution of Great Britain; British Association for the Advancement of Science; Sigma Xi; Engineers' Club; Franklin Institute of Pennsylvania; International Association for Advancement of Science, Art and Education; International Association for Testing Materials of Engineering; Institution of Engineers and Shipbuilders of Scotland; Kongl. Svenska Vetenskap-Academien; Naval Order U. S.; Nat. Geographical Society; American Meteorological Society; American Historical Society; New York Academy of Science; Oesterreichische Ingenieur und Architekten Verein; Order Loyal Legion of U. S.; Société des Ingénieurs Civils de France; Society of American Wars; Société d'Encouragement, etc., of France; Société Industrielle de Mulhouse; U. S. Naval Institute; Verein Deutscher Ingenieure; Washington Academy of Sciences.

The list of his publications are: Report of the Universal Exposition at Vienna (1873); History of the Steam Engine (1878); Report on Alloys (1878); Materials of Engineering (first volume, 1882; second volume, 1883; third volume, 1884); Stationary Steam Engines (1883); Development of the Philosophy of the Steam Engine (1884); Treatise on Friction and Lost Work in Machinery and Mill Work (1885); Steam Boiler Explosions in Theory and Practice (1877); Manual of Steam Boiler Design (1888); Hand-book of Engine and Boiler Trials (1890); Heat as a Form of Energy (1890); Reflections on the Motive Power of Heat (1890); Life of Robert Fulton (1891); Manual of the Steam Engine (1891).

Professor Thurston took great interest in the achievement and development of the steam turbine, but particular attention was paid to the Parsons-Curtis form.

He was a member of the Municipal Council of the city of Ithaca and took part in the discussions as to the practical advisability of the introduction of a voting machine to contribute both to ease and rapidity of the voting procedure and to the exactness of the count. Some of his comments will be found in a paper before the Society.

The debt which American Mechanical Engineering and the work of the technical school of this country owes to Dr. Thurston will probably never be adequately recorded. The results of pioneer work are often only the parts which are in evidence, the steps which have led to the result being obliterated or under-recorded.

Furthermore, the difficulty of the later steps is much less than that of the earlier process, and the results which cost so much to attain by the earlier process are frequently swept away by the improvements of the later day. Professor Thurston leaves behind him as his most striking monument his work as an author, his work in building up Sibley College and the admiring memory in the hearts of his students and professional associates.

Professor Thurston's contributions to the *Transactions* of this Society are as follows:

"On the Ratio of Expansion at Maximum Efficiency"; "Note Relating to the Proper Method of Expansion of Steam and Regulation of the Engine"; "Our Progress in Mechanical Engineering"; "On the Several Efficiencies of the Steam Engine, and on the Conditions of Maximium Economy"; "The Mechanical Engineer, His Work and His Policy"; "On a Newly Discovered Variation in the Effect of Prolonged Stress on Iron"; "Note Relating to Water Hammer in Steam Pipes"; "Pressures Obtainable by the Use of the Drop Press"; "A New Theory of the Turbine"; "Theory of the Sliding Friction of Rotation"; "Steam Boilers as Magazines of Explosive Energy"; "On the Theory of the Finance of Lubrication and on the Valuation of Lubricants by Consumers"; "On the Friction of Non-condensing Engines"; "A Note on Helical Seams in Boiler Making"; "The Systematic Testing of Turbine Water Wheels in the United States"; "Internal Friction on Non-condensing Engines"; "Proportioning Steam Cylinders"; "Large, and Enlarged Photo-

graphs and Blue-prints"; "On the Distribution of Internal Friction of Engines"; "On Variable Load, Internal Friction, and Engine Speed and Work"; "Philosophy of the Multi-cylinder or Compound Engine: Its Theory and Its Limitations"; "Hirn and Dwelshauver's Theory of the Steam Engine; Experimental and Analytical"; "Chimney Draft, Facts and Theories"; "Authorities on the Steam Jacket, Facts and Current Opinion"; "Steam Engine Efficiencies: the Ideal Engine Compared with the Real Engine"; "Technical Education in the United States"; "On the Maximum Contemporary Economy of High Pressure Multiple-expansion Steam Engines"; "The Theory of the Steam Jacket, Current Practice"; "Superheated Steam, Facts, Data and Principle"; "The Promise and Potency of High-pressure Steam"; "Multiple Cylinder Steam Engines, Effects of Variation of Proportions and Variable Load"; "Graphic Diagrams and Glyptic Models"; "Steam Engine at the End of the Nineteenth Century"; Reheaters in Multiple Cylinder Engines"; "Steam Turbine."

No. 1050.

MEMORIAL NOTICES OF MEMBERS DECEASED DURING THE YEAR.

JEREMIAH M. ALLEN.

On December 29, 1903, Mr. Jeremiah M. Allen, widely known throughout the United States as President of the Hartford Steam Boiler Inspection and Insurance Company, died at his home in Hartford, Conn. Mr. Allen was born on May 18, 1833, in Enfield, Conn.

Hartford will ever remember him as the first President of its Board of Trade, organized in 1888, and also for the work which he did while a member and President of the Board of Trustees of the Hartford Theological Seminary. He was directly or indirectly connected with nearly every financial or industrial institution in Hartford. Mr. Allen was a member of many scientific societies, among them The American Society of Naval Engineers, The American Association for the Advancement of Science, The American Historical Society, The Connecticut Historical Society, and this Society, of which he became a member in August, 1881.

The life work of Mr. Allen may be said to have been given to the company of which he was head, he having held the presidency since 1867. Even before that time he was greatly interested in insurance matters. He had for a number of years been a lecturer on insurance topics at Sibley College of Cornell University and at the Worcester Polytechnic Institute. He also founded the magazine called "The Locomotive."

JACOB NEFF BARR.

Mr. Jacob Neff Barr was born in Lancaster County, Pennsylvania, July 9, 1848. During his boyhood he lived on a farm and attended public school until the age of sixteen, when he began teaching and studying at the same time to fit himself for college. After teaching for some time he attended the Millersville, Pa., State Normal School, from which he was graduated in 1869,

having pursued both the scientific and normal courses there. He then entered Lehigh University, and was graduated as a Mechanical Engineer in the early '70s. After leaving Lehigh he entered the service of the Pennsylvania Railroad Company at West Philadelphia, remaining there one year, and was then transferred to Altoona, where he entered the drafting-room, and was soon after appointed Superintendent of the Wheel Foundry. While at Altoona he began his experiments with the Sand Flange and Contracting Chill Methods of casting car wheels, both of which methods he patented in 1883. During this year he accepted the position of Mechanical Engineer on the Chicago, Milwaukee & St. Paul road, with headquarters at Milwaukee, and two years later was made Superintendent of Motive Power. He remained with the C. M. & St. P. road in this capacity for a long time. In 1899 he accepted the position of Superintendent of Motive Power of the Baltimore and Ohio R. R., with headquarters at Baltimore, remaining there until the fall of 1901, then taking the position of Superintendent of Motive Power of the Erie R. R., with headquarters first at Binghamton, N. Y., and later at Meadville, Pa. In May, 1902, he left Meadville and the Erie road to take the position of General Superintendent of the C. M. & St. P., with headquarters at Chicago. One year later he was promoted to the position of Assistant to the President of the C. M. & St. P. road. He occupied this position until the time of his death. In the fall of 1903, after suffering from a severe attack of pneumonia, he developed a complication of heart and kidney trouble, and on the advice of his physician took a six months' leave of absence, going to California in January, 1904. He was not benefited by his stay in the warmer climate, however, and returned to his home in Libertyville, Ill., during the month of April. He rapidly grew worse after his return, and passed away at his home in Libertyville, Ill., on May 15, 1904. In addition to his inventions of the Sand Flange and Contracting Chill Methods of car-wheel casting, Mr. Barr made numerous other important inventions, having taken out sixteen patents, all of them covering improvements in either cars or locomotives. Mr. Barr was a prominent member of the Master Mechanics' and Master Car Builders' Associations, and had a thorough knowledge of the rules governing the interchange of freight cars.

For years his original ideas on car interchange impressed themselves on those interested in the revision of the rules at the Annual

Conventions, and the present code, which is largely based on the larger responsibility of the car owner for cost of repairs, is due in a great measure to Mr. Barr's labor for that principle.

He was also a member of the Western Railway Club—at one time President—and contributed valuable papers for its Proceedings. He was a man of exceptional executive power and of remarkable mechanical ability. He became a member of the Society in May, 1885.

ROBERT C. BLACKALL.

Mr. Robert C. Blackall was born in Albany, New York, in 1831, and had been actively connected with railroad work since 1850. He was a journeyman machinist in the Saratoga and Washington shops until 1853, machinist and gang foreman with the Hudson River Railroad until 1860, and successively advanced through other positions on the Delaware and Hudson or its constituents.

From 1870 to 1899 he was Superintendent of Motive Power and Machinery of the Delaware and Hudson. At the time of his death he was Consulting Mechanical Superintendent of the D. & H. Canal Company.

Mr. Blackall, besides being a member of this Society, which he joined in June, 1894, was also a member of the American Railway Master Mechanics' Association, of which he had been President, and the New York Railroad Club, having been President of the latter for several years. Mr. Blackall died at Albany, August 31, 1903.

ROBERT M. BLANKENSHIP.

Mr. Blankenship was born in Richmond, Va., May 6, 1866. His preparation for his life work was along general academic lines and in the Chemical Laboratory of the Iron and Steel Works. In 1888 he entered the Old Dominion Iron and Nail Works, as assistant to the Superintendent, and was ultimately made Superintendent. He connected himself with the Society at the Providence Meeting in 1891, and his death occurred during the early part of 1904.

COL. HENRY M. BOIES.

Col. Henry Martin Boies was born on the 18th of August, 1837, in Lee, Mass. He was of French-Huguenot origin, his ancestors migrating to Boston. He entered Yale College and graduated in 1859. He entered business life at once as a member of the firm of Silver & Boies, engaged in transportation on the Hudson River between Tivoli and New York. He at one time, while in

Chicago, joined the Ellsworth Zouaves, and laid the foundation of military skill and experience which afterwards proved of great service in the National Guard of Pennsylvania. He began his continuous and most successful career in 1865 at Scranton, Pa. Here he developed rapidly business, social and mental qualities which won him success, friendship and honor, and rounded out with symmetry an energetic life, filled with altruistic purpose and accomplishments. He started in the powder business as resident member of the firm of Laflin, Boies & Turck, which was merged in the Moosic Powder Company, of which he was President. It was allied to the Laflin & Land Power Company, of which he was an active director, which afterwards became connected with the Dupont interests, whose business parallels the life of the Republic. In the early days of his business he secured three patents on his invention of the powder cartridge, an ingenious device to diminish the hazardous risk of the use of powder by the miner, and to conserve his safety in several ways. While in this business he was called to the presidency of the Dickson Manufacturing Company, which took upon itself under his guidance a greatly enlarged plant and diversified output of manufacture for a wide and far extended territory. At this time his attention was called to the need of an improved steel-tired and steel wheel for railroads. His study resulted in three patents and the erection of a plant for their manufacture. After making it a success he sold the business to become united with the larger trust which competition made necessary in order to secure the more perfect output and larger pecuniary results.

He was a ready writer, and furnished articles for the newspapers and magazines on a variety of subjects. He was the author of two publications, one entitled "Prisoners and Paupers," the other "The Science of Penology," which have received commendation from high sources. These were the outgrowth of his work and study while a member of the Board of Charities of the State of Pennsylvania, an appointment which he first received from Governor Beaver, and which was followed up by reappointment by succeeding governors. He became a member of the Society May 14, 1890.

SIR FREDERICK JOSEPH BRAMWELL.

Sir Frederick Joseph Bramwell, first Bart., created in 1889, D.C.L., LL.D. and F.R.S., was born in London, March 7, 1818,

and was educated at the Palace School at Enfield. He served his first apprenticeship, commencing in 1834, with Mr. John Hague, and later became Chief Draftsman and Manager of large engineering works and in 1853 established himself as a Consulting Civil Engineer. He became an Associate Member of the Institution of Civil Engineers in 1856, a Member in 1862, and acted as President of that Institution in 1884. He was also an active member of the Institution of Mechanical Engineers, serving as President in 1874 and 1875. He was knighted in 1881, and in 1884 acted as Chairman of the Executive Committee of the Inventions Exhibition; was made President of the British Association for the Advancement of Science in 1888; from 1885-1900 served as Honorary Secretary of the Royal Institution of Great Britain, in honor of which services his bust, by the late Mr. Onslow Ford, has been placed in the anteroom of the lecture hall of that Institution. One of the latest great undertakings in which he was concerned was the South Wales Electrical Power and Distribution Company.

He was the recipient of many honors from scientific societies, and was elected a fellow of the Royal Society about 1873. He was particularly active and energetic in all his work, and was especially emphatic as to the dignity and responsibility of the engineer as owing it to the public that no unworthy schemes reached maturity through lack of exposure by competent engineers, going so far on one occasion as to publicly denounce a plan which he had been retained to advocate.

He was elected to Honorary Membership in this Society November 6, 1884.

WILLIAM PITT CANNING.

Mr. William Pitt Canning was born in Stockbridge, Mass., September 17, 1844, and died at Warren, Mass., September 17, 1903.

His education was received from his father, Edward W. B. Canning, and at the Stockbridge Academy. He prepared for college, but on account of his health gave up undertaking a college course, and went to Boston, where his father had been appointed Deputy Collector of the Port.

Soon after his arrival in Boston he enlisted, serving first in the Army of the Potomac, and later he was a member of Company K, 33d Regiment, Massachusetts Volunteers, which was destined for service in Louisiana, but in consequence of ill-health

he was invalided home in August, 1863, one year after his enlistment. After regaining his health, he served two and one-half years in the Charlestown Navy Yard.

While there, he was engaged in repairs on the "Kearsarge" just after its fight with the "Alabama." He then served the remainder of his four years' apprenticeship in North Andover and Lowell.

About March, 1871, he went to Warren, Mass., as chief draughtsman of the Knowles Steam Pump Company, remaining there, with a few months' intermission, until 1879, when he went to Lowell, becoming Mechanical Engineer for the Lowell Machine Shop, in which position he remained till 1897, when he went to Warren as Designing Engineer for the Warren Pump Company.

Prior to 1871, when at the Lowell Machine Shop, he was often employed to work out over the drawing-board, or to reduce to practice in metal the ideas of some inventor of textile machinery. During the period of his employment as Mechanical Engineer by the Lowell Machine Shop, from 1879 to 1897, he had charge, under the direction of the Superintendent, Mr. Charles L. Hildreth, of the design and manufacture of a variety of important textile machines. From time to time it became his duty to introduce into their manufacture a standard machine which had not before been manufactured by that corporation, or to change the design of some machine, for the purpose of removing some inherent defect, or of adapting it to some peculiar application. Sometimes the problem was to "Americanize" a machine of foreign origin. Such duties gave opportunity for, and required the exercise of, his inventive powers within the limitations which were fixed by the "standard" nature of output with which he was concerned. His designs and inventions had a stability and reliability, after he had tested and modified them to his satisfaction, which made it practically certain that they would also be continually satisfactory to those who applied the machines to the uses for which they were adapted.

He was a man who won and retained to an unusual extent the confidence, coöperation and esteem of his associates.

He was elected a member of the Society in May, 1890.

JAMES A. CONNELL.

Mr. James A. Connell was born in New York City April 4, 1850. He received his preliminary education in the public schools

of the city, and later completed a course in Cooper Union, receiving the degree of Mechanical Engineer from that institution.

At an early age he learned the machinist's trade, and worked as machinist and tool-maker in various shops in New York for several years, then, turning his attention more to steam engineering, he was appointed Chief Engineer of the American News Company's building. In 1881 he was appointed Chief Engineer of the New York Produce Exchange, the position which he held at the time of his death. He became a member of this Society in 1893, and always took a great interest in its welfare and was a regular attendant at most of the meetings.

Mr. Connell had a strong personality and made many firm and lasting friends among his business associates.

He died on January 2, 1904, from pneumonia, after an illness of only a few days.

RUSSELL WHEELER DAVENPORT.

Mr. Russell Wheeler Davenport died in Philadelphia on March 2, 1904, distinguished both as a metallurgist and a manager of men.

The services rendered by Mr. Davenport to the steel industry of this country while the manufacture of the higher grades of open-hearth steel was in its infancy have never been fully realized or appreciated by the members of our Society.

When in the early eighties the Ordnance Bureau of our Navy Department decided to equip the cruisers with modern high-power steel cannon, the specifications for the steel to be used in the tubes, jackets and hoops of these guns were made more severe than those adopted by any European government at that time. And the problem of making steel for successfully filling these specifications was solved by Mr. Davenport in the works of the Midvale Steel Co. at Philadelphia, without the aid or advice of any foreign metallurgist or company, and without even the advantage which might have been derived from the inspection of foreign steel works.

Even at this time, after twenty years of experience and with the aid of the best talent in this country and Europe, the manufacture of steel to successfully meet these specifications presents one of the most difficult of metallurgical problems.

The series of experiments systematically and carefully conducted by Mr. Davenport through a term of several years, which

ended in the production of gun steel at the Midvale works equal to any in the world, surely deserves to rank among the most notable metallurgical achievements of this country.

The difficulties and discouragements which Mr. Davenport met with and successfully overcame in this work can only be appreciated by those who were with him at the time.

Probably the reason that his work at Midvale never received the general recognition which it deserved was that soon after he had successfully solved the problem the Bethlehem Steel Company decided to go into the manufacture of ordnance on so large a scale that the eyes of all engineers were turned upon them, and the plant of the Midvale Company at that time remained comparatively so small that Mr. Davenport, the true pioneer in this field, was overshadowed.

It is a notable fact, however, that before the ordnance plant of the Bethlehem Steel Company was completed the services of Mr. Davenport were secured by them, and that under his direct supervision their furnaces were started so smoothly and successfully upon their work as to excite the admiration of the entire engineering fraternity.

It is needless to point out the far-reaching effect which the ability to accurately fill specifications for the higher grades of steel has had upon all branches of engineering in this country.

As a manager of men Mr. Davenport was as successful as in his metallurgical work. Although he was always a rigid disciplinarian his straightforward truthfulness, his tact, his kindly disposition and his untiring energy obtained for him the love and respect of all square men with whom he came in contact. And in his long and successful intercourse with the Army and Navy Departments at Washington he demonstrated beyond question the fact for which he always contended—that the policy of simple, direct truthfulness was the only one to adopt.

For three years before the purchase of the Bethlehem Steel Company by its present owners he successfully filled the place there of Superintendent of Manufacture. This was during the trying period of the reorganization of that company, and his position during this time can only be adequately described as that between the hammer and the anvil.

For the year before his death he was Vice-President of the Cramp Shipbuilding Company, and his management of their affairs was so successful that he was to have been unanimously elected president of that company at their next meeting.

Mr. Davenport was born in Albany, N. Y., on March 26, 1849. He entered the Sheffield Scientific School in 1868, and graduated in 1871; spent 1871 and 1872 as assistant to Professor Brush in the chemical laboratory, and then took a course in the Royal School of Mines at the University of Berlin. In May, 1874, he went to the Midvale Steel Company as chemist, and successively occupied the positions there of Assistant Superintendent, Superintendent and General Manager. In 1888 he accepted the position of Assistant Superintendent of the Bethlehem Steel Company. For a number of years he was the Second Vice-President of that company, and for three years before his resignation in 1901 he was Superintendent of Manufacture. At the time of his death, March 2, 1904, he was the Vice-President and Manager of the Cramp Shipbuilding Company of Philadelphia. He became a member of the Society in 1902.

CHARLES M. DAY.

Mr. Charles M. Day was employed by the Hopedale Machine Company of Hopedale, Mass. (the company was later merged with others into the Draper Company), served an apprenticeship, worked in various shops, notably those of Pratt & Whitney, Brown & Sharpe, and other high-grade places. He returned to Hopedale, and took charge of the tool-making department of the concern. Later he was given charge of the most important manufacturing department of the works, and still later was appointed General Superintendent, which place he held at the time of his death.

He was a member of the Board of Directors, and always filled the positions to which he was elected with entire satisfaction. He was an honest, able, upright man; of great executive ability, a fine mechanic and one whose influence was felt not only in his business, but in the administration of the affairs of government. His death was a great loss to his business associates, and a severe blow to the community in which he lived.

Mr. Day's death occurred in the latter part of February, 1903, at the age of about forty-three. He became a member of the Society in May, 1884.

ELIHU DODDS.

Mr. Elihu Dodds was born near Pittsburgh, Pa., in 1842. At the beginning of the War of the Rebellion he enlisted in Company

A, 5th Regiment, Excelsior Brigade, 74th New York Volunteers, and was color-bearer for about two years, winning a reputation for bravery possessed by few. At the close of the war he located at Indianapolis, Ind., and in 1866 entered the service of Chandler & Taylor, serving successively as pattern-maker, foreman of pattern shop, draughtsman, designer, assistant superintendent and, finally, superintendent. This position he held until stricken with that dread disease, tuberculosis, which caused his death on June 10, 1903, at Phœnix, Arizona. He was thorough and painstaking in all of his undertakings, and won the respect and confidence of all with whom he came in contact. He became a member of the Society May 31, 1887.

CHARLES F. ELMES.

Mr. Charles F. Elmes, President of the Chas. F. Elmes Engineering Works, Chicago, and one of the oldest and most prominent mechanical engineers of that city, died as a result of paralysis on January 10, 1904. Mr. Elmes designed the pumps and propelling machinery for the first Chicago fire-boat, and also designed the first fire-boat for the city of Milwaukee, both of them proving highly efficient. Mr. Elmes was born in Hallowell, Maine, December 1, 1845, removing to Wisconsin in 1858, and three years later to Chicago. Serving an apprenticeship in the machine shop conducted by his father, Carleton D. Elmes, he gained thorough practical knowledge as a machinist, engineer and draughtsman which contributed to his great success in the engineering field. His father and he then formed a copartnership under the name of Elmes & Son, which continued until the death of the elder Elmes in 1877. From 1877 to 1895 the business was conducted under the name of Charles F. Elmes. In the latter year the corporation was formed which has since been known as the Charles F. Elmes Engineering Works, his two sons, Carleton L. and Charles W., who for several years had been in their father's employ, being admitted as stockholders of the new company. Mr. Elmes became a member of the Society in November, 1883.

CLARK FISHER.

Mr. Clark Fisher was born at Levant, Maine, May 27, 1837. His preparatory studies were undertaken at the Academy at Trenton, N. J., and he entered the Rensselaer Polytechnic Institute at

Troy, N. Y., in 1854, and graduated in 1858 as Civil Engineer. He entered the United States Navy as Third Assistant Engineer May 3, 1859, became Second Assistant July 1, 1861, and First Assistant July 25, 1866, and was made Chief Engineer January 23, 1871. He left the service on account of the death of his father and the loss of his brother, who was killed at Fort Fisher, in order that some one of the family should carry on the business. His record of military service is as follows:

"Clark Fisher served in the navy from early in 1859 to 1872, in the various grades of his corps, through the War of the Rebellion, participating in the engagements at Whitehouse Landing; Pocotaligo in 1862; in the attack on Morris Island, Charleston harbor, including the bombardment of Fort Sumter; the attack on Fort Wagner and Stono Inlet in 1863; the advance up the James River in the attack on Howlett's, and the Dutch Gap Canal in 1864.

"He was taken prisoner at Magnolia Station, S. C., in 1862, but escaped the hardships of a prison life after being confined but one night. He was a brilliant engineer, and did some valuable work for the Navy Department at the Brooklyn Navy Yard in the elaborate and conclusive experiments he carried out at that station that established the value of oil as a fuel, and in originating devices for the economical combustion of this fuel, clearly demonstrating its value and its limitations, which are now a part of the records of the Navy Department.

"Mr. Fisher was highly esteemed by Commodore Isherwood, Chief of the Bureau of Steam Engineering of the Navy Department during the War of the Rebellion, for his ability in carrying out the experimental work at the Navy Yard and developing the science of modern steam engineering."

His father, Mark Fisher, established the Eagle Anvil Works, in Trenton, N. J., which Clark Fisher carried on after his father's death, and made it a great success. He was a member of the American Society of Civil Engineers, and joined this Society August 1, 1893. Clark Fisher died at Flushing, Long Island, December 31, 1903, after a severe illness, and his life was lengthened by the devotion of his wife, who had already nursed him faithfully through two severe attacks in preceding years. The Government detailed a naval escort for the funeral, and he was buried in Trenton, N. J., with naval honors.

DAVID ROSS FRASER.

Mr. David Ross Fraser was born at Berwick-on-the-Tweed, Scotland, on May 18, 1824, and came to the United States in 1848 (May), at the age of twenty-four years, where he was first employed in machine-shop work at Pittsburgh. He arrived in Chicago September 5, 1848, and his first employment there was with Gates & Hoag, which firm later became Gates & McHight. Mr. Thos. Chalmers, with whom he created the firm of Fraser & Chalmers twenty-three years later, was then in the employ of Gates & Hoag.

In April, 1850, he crossed the plains to California as a gold-seeker, but fever and ague drove him back in the same year, returning via Panama. However, he returned to California in 1852, seeing an opportunity for mechanics, and was employed in machine shops in San Francisco; but though gaining an exceptional reputation as a mechanic and earning extraordinary wages and with a fine prospect for advancement, ague again drove him back. He again returned to Chicago via Panama in 1853, and entered the employ of Scoville & Sons' Locomotive Works as foreman. These works stood on the land now occupied by the Union Depot at the corner of Canal and Adams Streets. Here he superintended the building of the first locomotive ever built in Chicago, and personally ran it by its own steam over the plank road then on Canal Street from Adams to Kinzie Street, delivering it to the Galena and Chicago Railroad, the first steam railroad in or out of Chicago.

In 1854 the Scoville & Sons' Works closed, and Mr. Fraser engaged with P. W. Gates & Co., where he built engines for some time by contract. As foreman, he continued with this firm until 1857, during which time he became a partner. Then came the organization of a new company, R. W. Gates & Co., operating under the title of the Eagle Works Manufacturing Co., at the corner of Canal and Washington Streets, and in which both Mr. Fraser and Mr. Thomas Chalmers were stockholders and acting Superintendents until 1872, when shortly after the great Chicago fire they withdrew and started the renowned business known as Fraser & Chalmers, which grew and prospered wonderfully under that name, and which name was retained until it was purchased in 1890 by an English syndicate.*

---

* This corporation has since been merged into the Allis-Chalmers Company.

Mr. Fraser went to England in 1890 and erected there the English works of Fraser & Chalmers, Limited, at Erith, on the Thames River, a short distance from London, and except for some months when he was called to Chicago to take charge of the Chicago factory during a period when it was nearly overwhelmed with orders for mining machinery, and which his brilliant executive ability in shop management overcame, he was engaged in erection of this English works and operating them until the spring of 1903.

In 1893 Mr. Fraser retired from active business, but was Vice-President and largest stockholder in the Chicago Portland Cement Company.

In November, 1851, he married Miss Lydia H. Scoville. Their golden wedding was celebrated in November, 1901, surrounded by their children and grandchildren.

Mr. Fraser has held an enviable reputation as a mechanical engineer and invented numerous mechanical devices. His genius was called into play to good purpose in the development and improvement of mining machinery, as will be recalled with pleasure by many engaged in the mining districts in every part of the world.

He possessed a most winning personality and made hosts of friends, his generous and kind yet firm disposition endearing him to the lowliest in his employ.

He died May 30, 1904, from a stroke of apoplexy, after twenty-four hours' unconsciousness following the attack.

He became a member of the Society May, 1886.

STEPHEN J. GEOGHEGAN.

Mr. Stephen J. Geoghegan was born May 10, 1836. At about the age of twelve he entered the employ of Mr. Thomas Carter, a manufacturer of gas fixtures, located on the Bowery near Grand Street, and remained four years.

He then entered the employ of the firm of Leak, Wood & Hunter, one of the pioneers in the manufacture of fittings for steam, water and gas supply.

Upon the death of Mr. Leak, in 1856, and the withdrawal of Mr. Wood from the firm, it then became the firm of Hunter & Kellar, and at this time Mr. Geoghegan was made Superintendent.

He remained with this firm until May, 1866, when, with Mr.

A. S. Cameron, he formed the firm of Cameron & Geoghegan, designing and constructing Heating and Ventilating Apparatus.

Later he successively allied himself with E. D. Slater, J. F. Riley, and T. A. Coles, until, in 1869, Mr. Charles J. Gillis became associated with him, forming the firm of Gillis & Geoghegan.

Mr. Geoghegan was a master mechanic in all that pertained to his work, and in the great advance and progress made in this line during his time he bore no small part developing, as well as successfully operating, his own business. His death occurred September 7, 1903.

He became a member of the Society May, 1887.

P. J. GREENWOOD.

Mr. Greenwood was born in England, May 19, 1840. He became a machinist apprentice and served the usual seven years with private instruction in drawing and mathematics. His mechanical work was on gun tools, for the making of pipe fittings, together with general work on locomotive and stationary engines.

Something over thirty years ago his attention was directed more and more to shoe machinery and their manufacture, and he followed this special line of work in connection with the Tredegar Iron Company of Richmond, until the time of his death.

He joined the Society at the time it met in his home city of Richmond, Va., in 1890. He died December 22, 1902.

JOHN HULETT.

Mr. John Hulett was born in Brooklyn, N. Y., March 14, 1874. At an early age he removed to Newburgh, N. Y., where he received his education in the public schools of that city, after which he entered the Newburgh Academy, graduating in 1892. The following year he took a post-graduate course at the Academy and conducted the Manual Training Shop Course. In September, 1893, he entered the Mechanical Engineering Course at Cornell and graduated in 1897. After leaving college he entered the draughting-room of the Metropolitan Street Railway Company, leaving there in 1898 to enter the employ of the Boston Rubber Shoe Company. In 1899 was connected with the Edison Electric Illuminating Company of Boston. Early in 1902 he accepted the position of Chief Engineer of the Monongahela Manufacturing Company of Monongahela, Pa. He became a member of the Society December 6, 1899.

JOHN HUMPHREY.

Mr. John Humphrey was born in Lyndon, Vt., October 12, 1834, but nearly all his life had been a resident of Cheshire County, having gone to Rindge when he was very young and afterwards living in Nelson, Harrisville, Marlboro, and, since 1856, in Keene, with the exception of two years in White River Junction, Vt. Mr. Humphrey's educational advantages were limited to such instruction as he could get in the public schools, but, having a thirst for knowledge and an excellent memory, he applied himself diligently and mastered such books as came within his reach, both on mathematics and mechanics, in which he was especially interested, and on general subjects. Later he studied mechanical engineering and hydraulics very thoroughly, and became an expert in the construction and designing of water wheels. He became a member of the Society May 15, 1889.

Mr. Humphrey started out at the age of twelve years to find employment and gain his own livelihood, his first engagement being in the woodenware business for about three years. Then, after clerking a year in a country store, he went into a woodenware mill in Nelson, and while there began making patterns for a machine he had invented, and later, after visiting a foundry in Harrisville to obtain information as to the shrinkage of metal, shaping and coring of patterns, etc., accepted a generous and encouraging proposition from Mr. Mainard Wilson to work in his shop and at the same time to build his machine. The machine was successfully built, and Mr. Humphrey remained at the shop until Mr. Wilson's death in 1854, the business afterwards continuing in his charge for a year. In 1855, being of age, Mr. Humphrey formed a partnership in Marlboro', where he engaged in business for a year as a machine-maker, etc., but the partnership not proving satisfactory he came to Keene and entered the machine shop of H. L. Haynes as an employee. He devoted his leisure time to the making of special machines, and in 1859, while setting up one of them, made an engagement at White River Junction, where he manufactured shoepeg and other machinery about two years, until the shop burned down. Mr. Humphrey then returned to Keene and took the shop of Mr. Haynes, beginning a business on his own account which has since continued successfully under the firm name of J. Humphrey, J. Humphrey & Co., and The Humphrey Machine Co.

Mr. Humphrey was one of the leading spirits in the erection of the Beaver Mills plant, and removed his machine shops there from Ralston Street on their completion. He made a great variety of wood-working machines from improved designs of his own, and in the line of clothespin and shoepeg machinery developed much that was entirely new. His shoepeg machinery is still sent to foreign countries where such pegs are in use to-day. His clothespin machines are also in use. He also made a great number of special machines designed by himself to solve complicated problems for others, and was very successful in that line of work. An excelsior machine was one of his later productions, and among other things designed and improved by him was a lumberman's log caliper for computing the contents in logs in cord or board measure.

The Humphrey Machine Company, of which Mr. Humphrey has always been the manager, was formed in 1873. He began making water wheels some years before that, but brought out his very successful I-X-L turbine about that time. There are a large number of these wheels in use in New Hampshire, Massachusetts and Connecticut, from which most satisfactory results have been obtained. The largest I-X-L wheel built by Mr. Humphrey, 100 inches in diameter, has been in use nearly twenty years by one of the big Lowell corporations. The X-L-C-R, a double wheel on a horizontal shaft, was designed later by Mr. Humphrey, and has also been most successful. In 1888 he purchased of the late Moses Ellis his iron foundry on Davis Street, which he has since carried on together with his machine shop.

For nearly six years he had been publisher of a monthly paper called "The Investigator," which, as its name indicates, treated of matters in mechanical and other lines for which there appeared occasion for investigation. A series of articles on "Water Power" treated of that subject in a manner that gave much information on points not generally understood, and during his illness many articles of interest relative to machinery have been published, while by rare will-power and persistency he also continued to direct his business, the details of which he always worked out himself.

Mr. Humphrey's death occurred on Monday, August 24, at the age of 68 years.

FRANK KEMPSMITH.

Mr. Frank Kempsmith began his mechanical career at the age of thirteen and became a highly-skilled machinist and tool-maker. He worked in various Eastern shops, including Garvin's in New York and Brown & Sharpe's, being noted for his skill, good judgment and conscientious work.

Later he went West, and was for a time a tool-maker in Springfield, Ohio. Afterwards he became Superintendent of the shops of Warner & Swasey, of Cleveland, the Lick Telescope being constructed there at that time. From there he returned to Springfield, where with two partners he started in business, building machine tools, and about sixteen years before his death moved to Milwaukee, where by himself he built up a business now known as the Kempsmith Manufacturing Company. Owing to ill-health he finally retired from business. His death occurred in Milwaukee on April 10, 1904. He became a member of the Society in May, 1886.

FREDERICK H. LAFORGE.

Mr. Frederick Henry LaForge's death occurred on August 1, 1904, at the Waterbury Hospital, where he had been under treatment since the first part of May. He had suffered for some months with paralysis.

Mr. LaForge was born in New Haven, Conn., May 4, 1835. His early school days were spent in his native city, and about 1852 he moved to Waterbury, Conn., where he took up his residence. As a mechanic, he worked in many of the local factories. In 1873 he was appointed Boiler Inspector for the Second Congressional District. He acted in that capacity from that time until he was taken ill.

Early in 1885 the users of steam in Waterbury took measures to establish a local company for boiler insurance on a "mutual" basis. Mr. LaForge was one of the first to aid in the movement. In 1886 he was chosen the first Chief Inspector of the Company, which was called the Connecticut Mutual Steam Boiler Inspection and Insurance Company. In 1894 he was re-elected to office.

In 1868 Mr. LaForge and W. Geddes patented a shafting machine; in 1869 he and G. E. Somers patented a screw press; in 1873 he patented covers for stop-clocks; in 1885 he and J. R. Smith patented a keyring tag, and in 1889 he and H. J. Barker patented a direct-acting steam engine.

In the early part of 1881 an association of stationary engineers was formed in Providence, R. I., and in 1884 Mr. LaForge was elected a Trustee of the organization.

He was elected a member of the Society in 1887.

PULASKI LEEDS.

Mr. Pulaski Leeds was born at Darien, Conn., in 1845. When thirteen years old he entered the shops of the New Haven and Hartford Railroad as an apprentice, and the whole of his subsequent work was done in railroad service. After finishing his apprenticeship, he followed a practice, very common and popular in those days, of becoming a locomotive fireman as a means of reaching the position of locomotive engineer. He ran a locomotive for about five years, and then returned to the repair shops as foreman. He rose by the usual steps of General Foreman and Master Mechanic, and in 1877 received the appointment of Superintendent of Motive Power of the Boston and New York Air Line. Two years later he went to a Western railroad, and after holding official positions on various railroads accepted the position of Master Mechanic of the Louisville and Nashville Railroad, with charge of the repair shops at Louisville, Ky. While holding that position Mr. Leeds became noted for the number of labor-saving appliances which he introduced, many valuable devices being of his own invention. He took a leading part in the introduction of pneumatic-driven appliances and other mechanism calculated to lighten the burdens and facilitate the operations of workmen.

In 1889 Mr. Leeds was advanced to the position of Superintendent of Motive Power of the company he had served so well as Master Mechanic. In that position he took a lead in developing the locomotive to the high power and efficiency the engine has now attained. While decidedly conservative in accepting innovations, Mr. Leeds was highly progressive on safe lines and had exceedingly keen perceptions of shortcomings in mechanical designs.

Mr. Leeds for years took an active interest in the various railroad mechanical associations. He was President of the American Railway Master Mechanics' Association for one year, and was a member of the Executive Committee of the Master Car Builders' Association. Mr. Leeds met an untimely death on July 8, 1903, by the pistol of a workman who had been refused a favor forbidden by the rules of the railroad company. Mr. Leeds became a member of the Society in May, 1902.

W. W. LINDSAY.

Mr. W. W. Lindsay was born in Philadelphia on December 25, 1865, and was educated in the public schools of this city. He graduated from the Central High School in June, 1882. During that summer he entered into an electrical contracting business on a small scale. In the beginning of 1883 he was employed for a short time by the George V. Cresson Company of Philadelphia. In January, 1887, he entered the employ of Gordon, Strobel & Laureau, Consulting and Contracting Engineers, and later became a partner in that company. Mr. Lindsay was well known among blast-furnace managers throughout the United States, having, during his connection with Gordon, Strobel & Laureau, charge of all field work. In 1891 he severed his connection with this company and became manager of the Barr Pumping Engine Company of Philadelphia. He remained with this company until the beginning of 1898, at which time he accepted a position as General Manager of the Deering Harvester Company in Chicago, each of these changes resulting in a position involving greater responsibility and ability. In the summer of 1898 he returned to Philadelphia, and engaged in the engineering and contracting business for himself under the name of W. W. Lindsay & Co., which business he carried on successfully up to the time of his death on November 12, 1902.

Mr. Lindsay was elected to the membership of the Society on May 17, 1892.

JOHN RUGAN MATLACK, Jr.

The Society has suffered during the year from the death of the elder Mr. Matlack and of the subject of the present sketch. He was born in Philadelphia on August 29, 1854, and after high school and polytechnic institute training, he entered the employ of the Pennsylvania Railway as rod-man. In 1875 he changed to the Philadelphia and Reading. In 1877 and for five years he was superintendent of R. W. Evans & Company. In 1882 he went into topographical work, and in 1883 took up sewerage systems and water works for the town of Olean, New York. In 1885, after a year of topographical work again, he connected himself with the firm of Riehle Brothers, of Philadelphia, and was their engineer for many years.

He became a member of the Society in 1892, and his death,

on January 10, '04, was the result of a railway accident. It was not considered to be a serious injury at the time, but resulted fatally after a short illness.

EDGAR HARRISON MESSER.

Mr. Edgar Harrison Messer was born at Reading, England, on 11th September, 1867, and was educated at Sidcot School, Somerset, and Bootham School at York. In 1885 he was apprenticed to Messrs. W. & J. Player, engineers, of Birmingham, and was employed there for three years in the fitting shops and for one year in the drawing office. In 1889 he proceeded to South Africa, and was employed for two years as an erector in various mines. Afterwards he was appointed Engineer-in-charge of the New Primrose Gold Mining Company's stamp mill, cyanide plant and hoisting works. At the beginning of 1893 he became Head Draughtsman to Mr. S. B. Connor, Consulting Engineer to the Consolidated Gold Fields of South Africa. With the exception of a visit to the Australian gold fields, extending over nine months (1896-7), he continued his connection with the Consolidated Gold Fields of South Africa, as Assistant Mechanical Engineer under Mr. J. B. Pitchford, and later under Mr. H. C. Behr, whose appreciation of his services both gentlemen were proud to acknowledge. Severing his long connection with the company in June, 1902, he joined the British Engineers' Alliance as Chief Engineer, where his thorough knowledge of every detail connected with the mining industry promised him a brilliant future. His career of usefulness was, however, cut short very soon after joining the Alliance. Contracting a severe cold, which developed into double pneumonia, he died in Johannesburg, on August 12, 1902, in his thirty-fifth year.

He was a man of many qualities, genial, open-hearted and a friend to all. His death caused quite a gloom in all circles of the community.

He became a member of the Society in May, 1901.

FAY DE VEAUX OLMSTED.

Mr. Fay De Veaux Olmsted was born in Denver, Colo., October 3, 1875. He died in Denver, October 3, 1903.

At the time of his birth his father was deeply interested in the study of milling methods as applied to the ores of Clear Creek County, Colo., and from the time of his earliest childhood Fay

displayed an inclination to be studious, especially along the line of mechanics. His physical constitution was never rugged, and as a result he was not permitted to enter school until eight years old. In his studies he was always far ahead of others of his age, even after his schooling had ended, and up to the time of his death, when he held the position of Assistant to the Chief Engineer of the Telluride Power Company, a concern operating four huge power-generating plants in Colorado, Utah and Montana. In all but name he was Chief Engineer, as his superior was absent most of the time, and his loss was felt keenly by the company for which he worked, by his associates and by his many warm friends with whom he had worked in various parts of the country.

In 1880 his parents, in order to overcome the effects of a high altitude on his nervous system, removed to California, living first in Los Angeles, later in Tehachapi and finally in San Jacinto, where, on May 8, 1891, his father died. His mother then returned to Denver with her three boys. Fay, after putting in two years at the East Denver High School and after exhibiting marked genius for engineering work, was sent to Ann Arbor to complete his preparatory course in the Ann Arbor High School, and thereafter to enter the University of Michigan.

In September, 1893, his mother died in Denver, and he was thrown upon his own resources with a small inheritance which was soon exhausted.

His first work after leaving college was in the railroad shops of Cleveland during the winter of 1897. Through the following summer he ran on the steamship "North Land" on the Great Lakes as Assistant Electrician. That autumn he was employed in the offices of the Northern Steamship Company at Buffalo, engaged in designing and draughting in connection with their repair work, and that winter went to New York to the employ of the Edison General Electric Company. He worked with this company as an Assistant Superintendent of Construction until March, 1899, when his health was shattered as the result of an attack of pneumonia which nearly ended fatally. In April following he came back to Denver in the hope of building up his weakened constitution, and might have succeeded had he continued to lead the out-door life into which he drifted in August of that year, when he obtained a position with the Telluride Power Company; but his worth was soon discovered by the officials of that company, and

gradually he was advanced in their service until he became the practical head of the engineering department. The indoor work which came with promotion was more than his frail physique could withstand, and his death followed soon after.

He became a member of the Society June, 1903.

JULIUS FR. PAJEKEN.

Mr. Julius Fr. Pajeken was born on September 16, 1843, in Bremen, Germany. He was the son of a well-known scholar and teacher, Clemens Albert Pajeken. He went to school in Bremen until he was sixteen, and then came to America, and served his time as an apprentice in the Morgan Iron Works, College Point, L. I. After this he returned to Germany and studied for four years in the Technical High School at Hanover. His first position after this work was in the firm—as a draughtsman—of Beyer, Peacock & Co., Manchester. In 1868 he entered the employment of Messrs. L. Schwartzkopf, Berlin, locomotive builders, and was with them for thirteen years. He was then an Engineer with the Sächsischen Maschinenfabrik, Chemnitz; and also for a short time in 1885 in the service of a large paper manufacturer in Leipzig. In the year 1888 he was appointed the Chief Engineer of the firm of Ludw. Loewe & Co., Actiengesellschaft, Berlin, and later on became one of the Managing Directors of the Company, a position which he held at his death on the 16th December, 1902, with special duties as a Manager of the Machine and Tool Works.

Few people realize what it means to create, as he did, a great machine tool works, built and equipped almost entirely along lines followed in a country over 3,000 miles away and carrying on its industries under entirely different social and political conditions. Mr. Pajeken took a special interest in young men, and many of them realize themselves to be indebted to him for wise counsel and substantial assistance. He was a man whom it was a privilege to know. He had an astonishing acquaintance with the English tongue, speaking it as fluently and idiomatically as an Englishman, a fact which no doubt assisted him largely in making friends in England and America.

He became a member of the Society in December, 1900.

ALEXANDER POLLOCK.

Mr. Pollock connected himself with the Society at its San Francisco Meeting in 1892. He was born in New York City, October

6, 1840, and, after a preparation in the public schools of the city and college of the city of New York, entered the Delamater Iron Works as apprentice in the machine shop, from 1857 to 1861.

He worked in the drawing-room with the late Capt. Ericsson as his chief from 1862-1866, the interval from April, 1861, to August, 1862, being given to service in the navy as acting Assistant Engineer. He served also as Superintending Engineer for Mr. John Baird of the Cromwell Steamship Company.

During the last thirty-five years of his life, he had been in business supplying machinery and acting as contractor for its erection. Mr. Pollock's death occurred on September 3, 1904.

CLYDE FITZ RANDOLPH.

Mr. Clyde Fitz Randolph was born at Salem, W. Va., in 1875. After graduating from Salem College, he served for a time as an apprentice in Plainfield, N. J., and in 1897 entered the Mechanical Department of the West Virginia University, from which he graduated in 1900. The following year he studied at Sibley College, Cornell University, receiving the degree of Mechanical Engineer in 1901. After leaving Cornell, he accepted a position with the Gray-Blaisdell Machinery Company, Bradford, Pa., and soon was promoted to the position of Chief Draughtsman, which position he held until September, 1902, when he resigned to accept the position of Instructor in Mechanical Engineering at the West Virginia University. His work as an instructor in machine design was of such high order that at the end of one year's service he was made Assistant Professor of Mechanical Engineering and placed in charge of the instruction in machine design and drawing. In addition to his work mentioned above, Professor Randolph had designed a number of gas engines, air compressors and special machines for various manufacturing concerns. He became an Associate Member of the Society in June, 1903. By his death the faculty of the West Virginia University loses one of its most able and popular members. He died at Morgantown, W. Va., May 16, 1904.

GEORGE RICHMOND.

There are few men who have been more prominent in the growth and development of the refrigerating industry in the United States than George Richmond, whose sudden death on June 4, 1904, came as a shock to his friends, business associates

and to the many who knew him by reputation. Ever since 1882 Mr. Richmond had been closely identified with manufacturing interests in the United States, not only as a patentee and manufacturer of refrigerating machinery, but also as a pioneer in the introduction of oil and gas engines for the development of power. For more than fourteen years he had been intimately connected with the De La Vergne Machine Company, formerly known under the name of the De La Vergne Refrigerating Machine Company, where his services were so highly appreciated that his counsel and advice were sought in almost every branch of the company's business. For years he was Consulting Engineer, not only in the refrigerating department, but also in the construction of other machinery. His practical knowledge made his advice invaluable in the sales department, which he supervised for several years previous to his death.

Mr. Richmond was an Englishman, and came to the United States in 1880. Soon after arriving in this country he took out several patents for refrigerating machinery and went into the manufacture of these machines under the firm name of Wood & Richmond in 1882. In 1891 Mr. Richmond formed a connection with the De La Vergne Refrigerating Machine Company as Expert Refrigerating Engineer, and held this position until 1894, when, upon the death of George F. Meyer, the Consulting Engineer of the Company, he succeeded to that office.

In 1895 a new department was created by the De La Vergne Company, under the name of Oil Engine and Motor Wagon Department. Prior to the opening of this department Mr. Richmond was sent to Europe to find the best type of gas or oil engine to be used in the operation of ice machines. After a careful investigation, four representative types of kerosene engines were selected and sent to his company to be tested. After careful experiment under his supervision the company decided to adopt an engine called the Hornsby-Akroyd Oil Engine, which had first been built in Grantham, England, by Hornsby & Sons, Limited. The efficiency of these engines proved so satisfactory that they have been used not only in the operation of ice machines but for furnishing power for other machines. They have also been extensively used by the United States Government in several of its departments. Incidentally, it may be mentioned that Mr. Richmond long ago recognized the importance of the use of motor vehicles and gave considerable attention to the subject.

Recognizing that the time was not far distant when gas would supersede steam in the development of power for large engines, Mr. Richmond, as Consulting Engineer for his company, again went to Europe in 1901. After a careful inspection of gas engines in use in Europe, he returned to this country, and in his report strongly recommended that the company acquire the right to manufacture and sell the Korting Double-Acting Two-Cycle Gas Engine. His recommendation was accepted and the company immediately proceeded to acquire the right and entered into the manufacture of these engines.

The scientific and practical services which Mr. Richmond had rendered to his company induced it to further call upon him, and he was placed in charge of the sales department, which under his direction was ably administered until the day of his death.

Since 1889 Mr. Richmond had been an active member of the American Society of Mechanical Engineers, and was one of the managers from 1897 to 1900. He contributed several most valuable papers, which were read before the Society, among which the following may be mentioned: "Notes on the Refrigerating Process and Its Proper Place in Thermodynamics," "The Refrigerating Machine as a Heater," "Thermodynamics Without the Calculus," "A Simple Calculating Machine." He was also appointed on a committee of the Society to establish a standard for conducting engine tests, as his research and intimate knowledge of gas and oil engines made him particularly fitted to aid the Society in the determination of this important question.

Mr. Richmond was a member of the Engineers' Club, and lately had been one of the moving spirits in the organization of a Society of Refrigerating Engineers.

For the last few years he had been engaged in the invention and perfecting of devices and appliances which would reduce the cost of the manufacture of ice. One of the most important works which has been left incomplete by his death is a treatise on the subject of refrigeration, which he had hoped to have published in the near future.

JOHN A. ROCHE.

Mr. Roche has the interesting prominence of being one of the several members of the Society who have taken active part in political achievement by becoming incumbents of responsible elective offices. Mr. Roche served the City of Chicago, Ill., as its Mayor, after fifteen years of service in business relations as Man-

ager of the Chicago house of J. A. Fay & Company, manufacturers of wood-working machinery.

Mr. Roche entered mechanical engineering lines by becoming an apprentice in the Allaire Works at New York, which he completed in 1864. From 1864-1868 he was engaged in steam engineering in Boston, in connection with J. R. Robinson, during which time he designed and superintended the construction of steam plants of various kinds. In 1868 he came to the city of Chicago, was subsequently one of the proprietors and superintendent of the Reliance Works, making a specialty of wood-working machinery. From this relation he entered the life-long connection with J. A. Fay & Company, acting as designer and contractor for the equipment of car and machine shops, and in the design of special machinery to meet the requirement of this type of establishment.

Mr. Roche allowed his membership to lapse after his term as Mayor had turned his interests away from mechanical matters, but he resumed his membership in August, 1902. His first connection was at the New York Meeting in 1886. He died February 10, 1904.

JOHN RUGGLES SLACK.

Mr. John R. Slack, until recently Superintendent of Motive Power of the Delaware and Hudson R.R., died of tubercular meningitis in New York City, August 1, 1904. Mr. Slack was born June 24, 1863, at Fishkill-on-Hudson, New York. He attended Columbia College, graduating there in 1884, and two years later graduated from Stevens Institute of Technology.

He was with the New York Central and Hudson River Railroad in the Motive Power Department, until 1888, when he was appointed Mechanical Engineer for the Central Railroad of New Jersey. In July, 1889, he was made assistant superintendent of motive power of the Delaware and Hudson, and in January, 1902, was advanced to Superintendent of Motive Power. At the time of his death Mr. Slack occupied the position of Assistant to the General Superintendent of the last mentioned road.

He became a member of the Society, December 6, 1899.

HENRY I. SNELL.

On the 20th day of October there passed away, in Philadelphia, one of the most faithful attendants upon the recurring meetings of the Society. Mr. Snell began his professional career previous to the widespread of the technical school, and began his profes-

sional work with a Civil Engineer, and under private instruction, in 1852. In 1855 he entered the service of the City Engineer of Lowell, Mass., and later, in 1857, attached himself to the proprietors of locks and canals of the Merrimack River under Mr. James B. Francis. For ten years, from 1863-1873, he was Mechanical Engineer and draftsman with the Allen Works of Boston. In the latter year, he connected himself with the house of B. F. Sturtevant, with whom he remained either in Boston or in Philadelphia, with a year's intermission of practice under his own name, until the time of his death.

Mr. Snell will be remembered as one of the participants in the excursion to England, in 1889, and as a delighted partaker in all the fun that attached to that enjoyable trip.

WILLIAM WALLACE.

Mr. William Wallace died in Washington, D. C., May 20, 1904, in the eightieth year of his age. He was born in England in 1825; he early came to this country with his father and established the firm of Wallace & Sons at Ansonia, Conn., which soon became one of the leading manufacturers of copper and brass alloys in the United States. Becoming associated with Prof. Moses G. Farmer, they began the manufacture of a compound telegraph wire, consisting of a steel core and an electrotyped copper covering, thus giving conductivity and strength, combined with lightness. In 1876, at the Centennial Exhibition, Mr. Wallace brought out the Farmer-Wallace Dynamo Machine, with which the buildings were successfully lighted, being the earliest general electric lighting in this country. A year or two later he devised a plate arc lamp for use with this machine, by means of which a number of arc lights could be placed in series on the circuit, thus originating the series method of arc lighting now so common. Mr. Wallace was greatly interested in scientific questions and had fitted up a laboratory in his house for experimental purposes. He constructed an induction coil of large size, which at that time was unequalled in this country. He was regarded as one of the leading authorities in the United States on the alloys of copper, zinc and tin.

Mr. Wallace was genial and agreeable in manner and willing to assist others in every possible way. For some years he had lived quietly in Washington, spending his time in working with his microscope, in the use of which he had become expert.

Mr. Wallace became a member of the Society in November, 1890.

JOHN Q. WRIGHT.

Mr. John Q. Wright, a veteran in the machine tool business, passed away with death by his own hand on the 16th of October, 1903. He was for a time General Superintendent of the works of the Putnam Machine Company, and for many years their representative in charge of the New York office. He joined the Society at its Pittsburg Meeting in 1884.

Index photographed at the beginning for the convenience of the microfilm user.